Industrial Dehumidification:

Water Vapor Load Calculations and System Descriptions

Accurately calculating water vapor loads in industrial environments helps size and select systems with minimal operating costs

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The need to control the amount of water vapor in the air is felt in all industrial, commercial, and institutional facilities. Humidity control is important to human health and comfort. Humidity control also improves the reliability of equipment, production processes, and materials by controlling static electricity, corrosion, and other factors. The purpose of this article is to provide information on energy usage, energy calculations, and operating energy costs to help engineers evaluate industrial dehumidification systems. I use mass-flow analysis, adiabatic mixing, and thermodynamics to evaluate the air- and water-vapor mixture as it travels through each component of the dehumidification system. This procedure produces an in-depth analysis of the sensible and latent heat energy flows. This analysis is also meant to help the engineer with the concepts of heat of condensation, re-

TABLE 1—Types of dehumidification systems.

Type	Common inlet air flows	Min. discharge air dew point temp.	Min. discharge air moisture* @ 29.921 Hg atm. pressure
1) DX cooling	50 to 50,000 cfm	42 F normal 35 F special design minimum (or lower)	39.45 29.92
2) Portable DX	150 to 330 cfm	40 F 23 F with defrost control & reduced capacity	36.48 18.19
3) Chilled water	500 to 50,000 cfm	40 F 37 F with 5 F approach on coil temp.	36.48 32.41
4) Chilled brine/glycol	500 to 50,000 cfm	32 F (below 32 F with frosted coil or defrost control)	26.51
5) Liquid desiccant	750 to 84,000 cfm	-80 F with LiCl Typical LiCl: 15 F (with 45 F CW) 25 F (with 55 F CW) 32 F (with 65 F CW) 40 F (with 75 F CW) 48 F (with 85 F CW)	0.063 12.89 19.78 26.51 36.48 49.68
6) Dry desiccant rotating bed	500 to 20,000 cfm	5 F	8.231
7) Dry dsiccant multiple vertical bed	500 to 25,000 cfm	-30 F	1.429
8) Dry desiccant rotating wheel	10 to 40,000 cfm	-80 F	0.063
9) Portable desiccant	200 to 350 cfm	5 F	8.231

Note: CW = Chilled water or cooling tower water
*Grains of water vapor per lb of dry air

activation heat, heat dump back, and desiccant heat.

Types of dehumidification systems

Some of the types of equipment and their moisture-removal capabilities are illustrated in Table 1. Moisture can be removed from the air by cooling it below the dew-point temperature so condensation occurs by air-to-air heat exchangers, which bring in dryer outside air, or by chemical methods. Chemical dehumidification is carried out through the use of sorbent materials, which are solids or liquids that can extract moisture from the air and hold it. There are two classifications of sorbents:

- Adsorbents—which do not experience a phase change. Moisture is deposited on the surface of the dry desiccant. Most adsorbents are solids.
- Absorbents—which change physically, chemically, or both during the sorption process. Most absorbents are liquids or solids that become liquid as they absorb moisture.

Portable dehumidifiers

When most people think of portable dehumidifiers, they think of equipment that is used in the home, but this is no longer true. These systems are being used in many commercial and industrial applications. A few examples of their uses are: indoor pools, cleaning and restoration, locker rooms, pump stations, libraries, restaurants and bars, film and tape storage, bakeries, well houses, and canning plants.

The types of portable equipment available are direct expansion (DX) reheat, dry desiccant, and air-to-air heat exchangers—to name a few. The water-removal capacities of these systems at ANSI B149.1 inlet air conditions (80 F and 60 percent RH) are as follows:

- DX reheat: 0.65 to 4.35 lb per hr; 1.88 and 12.50 gal per day, respectively.
- Dry desiccant: 6.26 lb per hr; 18 gal per day.
- Air changers: 15.30 lb per hr; 44 gal per day. (Building air is exhausted through an air-to-air heat exchanger that brings in dry outside air. Removal capacity is based on outside air at 0 F and 60 percent RH.)

Table 2 is a summary of more than 30 portable dehumidifiers reviewed for this article. The table illustrates both the water-removal capacity and the operating cost of each system. Notice that the energy-inefficient systems use up to four times more energy than the efficient systems. Table 3 presents the variation of water-vapor removal and the operating cost at various inlet air conditions. This table was based on one of the energy-efficient models. Some DX-reheat units will

TABLE 2—Capacity of portable dehumidifiers

Equipment	Type	Water removal[1]	Operating cost[2]
1. High Efficiency	DX - Reheat	13.20 gal/day or 4.60 lbs/hour	$7.96/1000 lbs. W.V.
2. High Efficiency	DX - Reheat	25.00 gal/day or 8.70 lbs/hour	$8.62/1000 lbs. W.V.
3. Low Temperature	DX - Reheat	18.00 gal/day or 6.4323 lbs/hour	$11.85/1000 lbs. W.V.
4. Conventional	DX - Reheat	5.0 gal/day or 1.7385 lbs/hour	$26.19/1000 lbs. W.V.
5. Conventional Low Efficiency	DX - Reheat	3.125 gal/day or 1.0865 lbs/hour	$31.75/1000 lbs. W.V.
6. Commercial	DX - Reheat	10.00 gal/day or 3.4770 lbs/hour	$18.85/1000 lbs. W.V.
7. Commercial	DX - Reheat	5.625 gal/day or 1.9558 lbs/hour	$35.99/1000 lbs. W.V.
8. Low Temperature	DX - Reheat	2.625 gal/day or 0.9216 lbs/hour	$57.65/1000 lbs. W.V.
9. Desiccant	Desiccant	17.0 gal/day or 5.9108 lbs/hour	$27.99/1000 lbs. W.V.
10. Restoration Dry	DX - Reheat	18.0 gal/day or 6.4323 lbs/hour	$11.85/1000 lbs. W.V.

Note: [1] Water removal is at ANSI B149.1. Inlet air conditions of 80F and 60 percent RH relative humidity.
[2] Operating cost is based on an electric rate of $0.06 per kwh

TABLE 3—Capacity of portable dehumidifier at various inlet air conditions

Tdb	Tdp	RH	W*	Removal	Operationg cost
1. 50°F	36.53°F	60%	31.73	3.16 gal/day	No Data
2. 50°F	40.47°F	70%	37.06	5.75 gal/day	No Data
3. 50°F	43.94°F	80%	42.40	8.34 gal/day	No Data
4. 60°F	41.20°F	50%	38.14	4.03 gal/day	No Data
5. 60°F	45.97°F	60%	45.85	8.13 gal/day	$8.07/1000 lbs. W.V.
6. 60°F	50.08°F	70%	53.58	9.49 gal/day	$7.64/1000 lbs. W.V.
7. 60°F	53.70°F	80%	61.34	11.88 gal/day	$6.68/1000 lbs. W.V.
8. 70°F	44.49°F	40%	43.32	4.31 gal/day	$15.20/1000 lbs. W.V.
9. 70°F	50.42°F	50%	54.28	8.34 gal/day	$8.69/1000 lbs. W.V.
10. 70°F	55.39°F	60%	65.30	10.38 gal/day	$7.98/1000 lbs. W.V.
11. 70°F	59.68°F	70%	76.37	12.66 gal/day	$7.50/1000 lbs. W.V.
12. 70°F	63.45°F	80%	87.50	14.13 gal/day	$6.84/1000 lbs. W.V.
13. 80°F	45.72°F	30%	45.42	4.31 gal/day	No Data
14. 80°F	53.45°F	40%	60.77	7.48 gal/day	$12.69/1000 lbs. W.V.
15. 80°F	59.62°F	50%	76.22	10.07 gal/day	$9.60/1000 lbs. W.V.
16. 80°F	64.79°F	60%	91.79	13.25 gal/day	$7.94/1000 lbs. W.V.
17. 80°F	69.26°F	70%	107.47	14.96 gal/day	$7.73/1000 lbs. W.V.
18. 80°F	73.97°F	80%	123.25	16.13 gal/day	$7.60/1000 lbs. W.V.
19. 90°F	54.35°F	30%	62.84	6.33 gal/day	No Data
20. 90°F	62.38°F	40%	84.19	9.49 gal/day	No Data
21. 90°F	68.80°F	50%	105.75	12.37 gal/day	No Data
22. 90°F	74.18°F	60%	127.51	14.96 gal/day	No Data
23. 90°F	78.83°F	70%	149.50	16.11 gal/day	No Data
24. 90°F	82.93°F	80%	171.69	17.54 gal/day	No Data

Note: This particular unit is a high capacity energy efficient unit. Tdb is dry bulb temperature. Tdp is dew point temperature. RH is relative humidity. Operating cost is based on an electric rate of $0.06 per kwh
*Grains of water vapor per lb of dry air

begin to form frost on the cooling coils when the inlet air dry bulb goes below 60 to 65 F. If the unit has frost control, it will experience significant capacity loss due to the frost-control operation. Two typical approaches to frost control are listed below:

- A temperature-sensing thermostat diverts the hot refrigerant gas through the evaporator coil until the ice is melted.
- An automatic de-ice sensor shuts the compressor off when evaporator-coil temperature approaches freezing. The fan continues moving warm air across the coil to defrost it.

Large industrial dehumidifiers

This section of the article is a detailed review of three common types of industrial dehumidifiers: direct expansion, dry desiccant, and liquid desiccant. Each system has an air intake of 30,000 cu ft per min at 70 F and 56 grains of water vapor per lb of dry air. Also, each system was required to dry the air down to 35 or 36 grains of water vapor per lb of dry air. I worked with several dehumidification companies to size the systems and determine all energy consumption. Using the brake horsepower provided by the companies, I sized the motors and calculated the electrical consumption (KW per hr) illustrated in this article. Operating costs were then developed using \$0.06 per KWH for electricity and \$5.00 per 10^6 Btu for natural gas. The operating costs were then developed into operating cost per 1000 lb of water vapor removed. In some cases, I left out proprietary information on the systems at the request of the manufacturers.

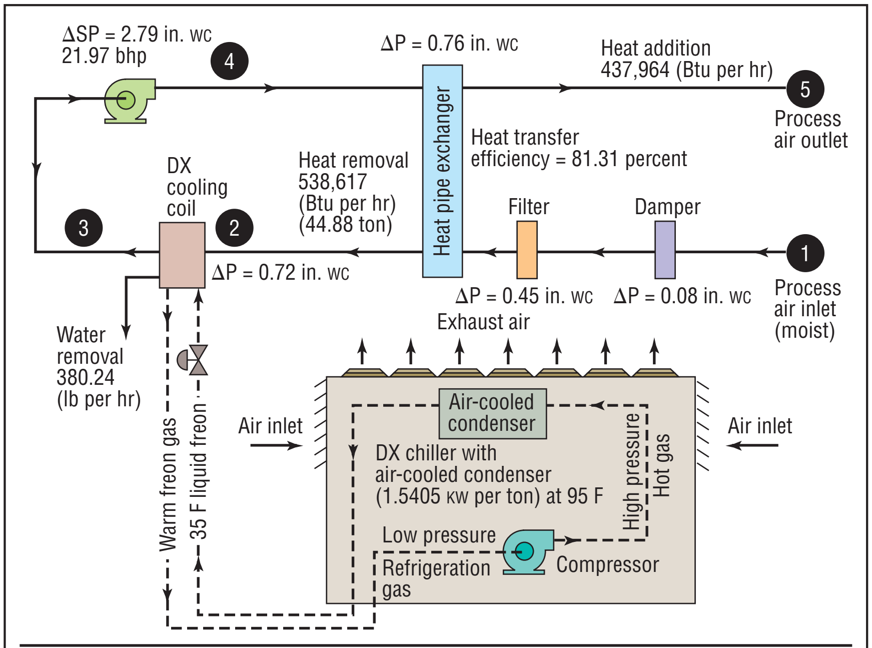
Each industrial dehumidifier includes detailed air- and water-vapor mass-flow analysis, psychrometrics, thermodynamics, and adiabatic mixing (Figs. 1 to 3) to help the readers understand the energy flows. The psychrometrics and thermodynamics used in the diagrams follow the principals of Zimmerman and Lavine. The air- and water-vapor flows are also illustrated in acfm (actual cu ft per min) and the dehumidification industry dscfm (dry standard cu ft per min). I used dscfm to avoid confusion with the fan industry scfm.

Cooling-based dehumidification

Moisture can be removed from the air by cooling the air below its dew-point temperature. This can be achieved through the following systems:

- Chilled water, glycol, or brine coil system
- DX cooling coil system
- Chilled water air-washer system

The first two systems accomplish dehumidification by passing the air through a cooling coil with a coil-surface temperature below the dew point of the air. Water va-



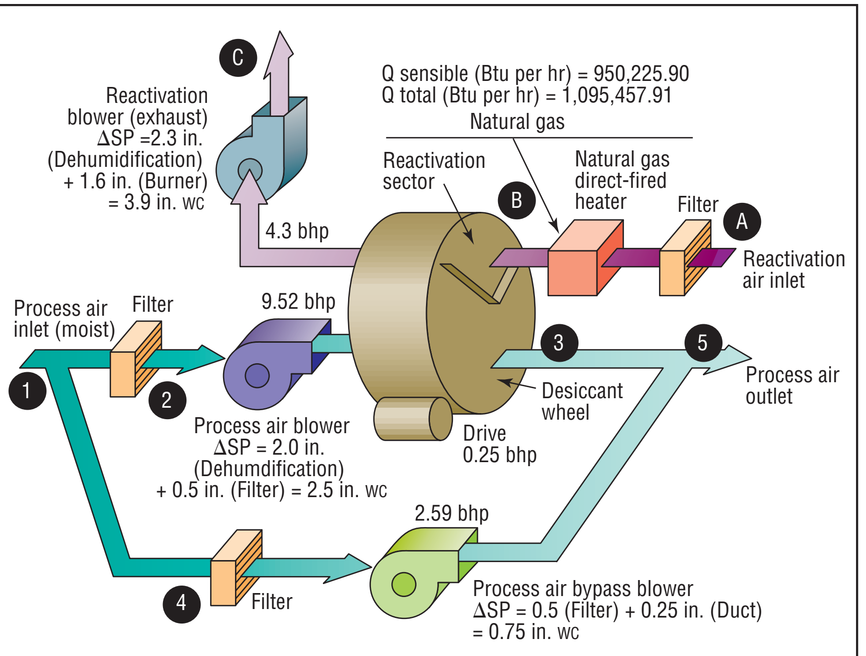
Process air anaylsis	1	2	3	4	5
Dry bulb	70 F	53.40 F	39.73 F	42.73 F	56.3 F
Wet bulb	58.8 F	52.17 F	39.73 F	41.29 F	47.90 F
Dew point	51.26 F	51.26 F	39.73 F	39.73 F	39.73 F
Relative humidity	51.56%	92.86%	100%	89.61%	53.95%
W (grains WV per lb dry air)	56	56	36	36	36
acfm	30,000	29,059.79	28,157.15	28,326.30	29,091.51
dscfm	29,574.27	29,574.27	29,574.27	29,574.27	29,574.27
M (lb dry air per hr)	133,084.56	133,084.56	133,084.56	133,084.56	133,084.56
M (lb WV per hr)	1064.68	1064.68	684.44	684.44	684.44
Q sensible (Btu per hr)	1,215,006	684,051	246,976	342,866	776,787
Q latent (Btu per hr)	1,162,977	1,155,315	738,627	739,522	743,565
Q total (Btu per hr)	2,377,983	1,839,366	985,603	1,082,388	1,520,352

Coil heat removal (information shows what occurs between Points 2 and 3)
ΔQ sensible (Btu per hr): 437,075
ΔQ latent (Btu per hr): 416,688
ΔQ total (Btu per hr): 853,763 (71.147 ton)
Water vapor removed (lb per hr): 380.24

1 DX with heat pipe dehumidifier.

por condenses on the coil surfaces. The amount of moisture removal depends on how cold the air can be chilled. The lower the temperature, the drier the air. The chilled water air-washer system also cools the air below its dew point by using water that is colder than the dew-point temperature. The water vapor in the air condenses on the water spray or the nearest surface. In this case, the use of colder water results in greater dehumidification.

The type of system chosen for illustration in this article is the DX cooling coil system. The basic components of this mechanical refrigeration system are an evaporator coil, compressor, condenser, and throttling valve (or expansion valve). The system uses a refrigerant that enters the evaporator coil (cooling coil) in a liquid state. The refrigerant evaporates inside the coil and, in doing so, absorbs heat from the process air moving through the coil. It then leaves the coil in the form of a gas. The compressor takes the cold vapor from the evaporator and compresses it to a hot gas at high pressure. When the refrigerant leaves the compressor, it is still a gas but at a much higher pressure (five to ten times greater) and a much higher temperature. The hot refrigerant gas is then pushed through a condenser (in this case, an air-cooled condenser) where the hot gas is cooled and condensed into a liquid by some substance, usually air or water. The refrigerant then flows from the condenser as a high-pressure liquid through the expansion valve. As the liquid passes through the valve, its pressure is suddenly decreased to the pressure in the evaporator coil. At the same time, the temperature of the liquid refrigerant drops down from the warm condenser temperature to the cold evaporator temperature. This occurs because a small amount of liquid suddenly flashes



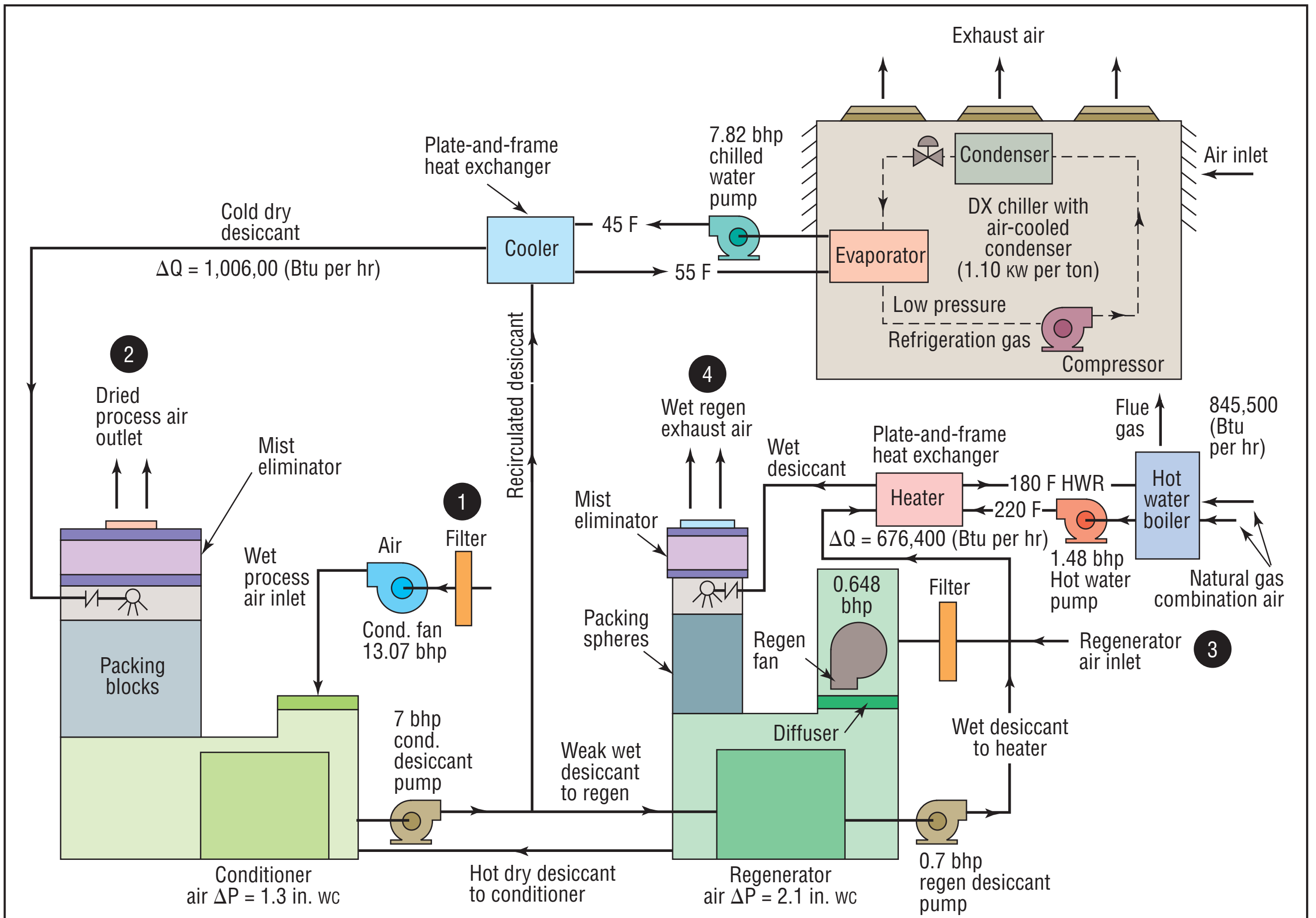
Process air anaylsis	1	2	3	4	5
Dry bulb	70 F	70 F	115 F	70 F	93.64 F
Wet bulb	58.80 F	58.80 F	65.64 F	58.80 F	62.54 F
Dew point	51.26 F	51.26 F	19.63 F	51.26 F	38.96 F
Relative humidity	51.56%	51.56%	3.62%	51.56%	14.98%
W (grains WV per lb dry air)	56	56	15.8	56	35
acfm	30,000	15,728.59	16,909.30	4,271.41	31,189.04
dscfm	29,574.27	15,505.38	15,505.38	14,068.89	29,574.27
M (lb dry air per hr)	133,084.56	69,774.41	69,774.41	63,310,15	133,084.56
M (lb WV per hr)	1064.68	558.20	157.49	506.48	663.97
M (lb per total hr)	134,149.24	70,332.61	69,931.90	63,816.63	133,748.53
Q sensible (Btu per hr)	1,215,006	637,011	1,393,584	577,995	1,971,579
Q latent (Btu per hr)	1,162,977	609,733	175,060	553,244	728,304
Q total (Btu per hr)	2,377.983	1,246,744	1,568,644	1,131,293	2,699,883

(The following information shows what occurs between Points 2 and 3.)
Heat of condensation (Btu per hr): 434,673
Reactivation system sensible heat leakage (Btu per hr): 321,900
Water vapor removed (lb per hr): 400.70

Reactivation air analysis	A	B	C
Dry bulb	90 F	293 F	130 F
Wet bulb	78.22 F	113.75 F	104.96 F
Dew point	74.06 F	81.24 F	100.76 F
Relative humidity	59.77%	0.88%	43.71%
W (grains WV per lb dry air)	127.0	162.25	308.07
acfm	4583.44	6310.45	5103.46
dscfm	4285	4274.78	4274.78
M (lb dry air per hr)	19,282.55	19,236.55	19,236.55
M (lb WV per hr)	349.84	445.89	846.59
M (lb per total hr)	19,632.39	19,682.44	20,083.14
Q sensible (Btu per hr)	268,784	1,219,010	453,895
Q latent (Btu per hr)	385,151	530,383	946,355
Q total (Btu per hr)	653,935	1,749,393	1,400,250

(The following information shows what occurs between Points B and C.)
Sensible heat loss total (Btu per hr): 765,115
Heat of desorption (Btu per hr): 443,215
Reactivation heat leakage to process air (Btu per hr): 321,900
Water vapor pick-up (lb per hr): 400.70

2 Rotating dry desiccant dehumidifier.



Process air anaylsis	1	2	3	4
Dry bulb	70 F	55 F	95 F	160.1 F
Wet bulb	58.80 F	47.00 F	78.40 F	132.12 F
Dew point	51.26 F	39.01 F	72.42 F	129.97 F
Relative humidity	51.56%	55%	48.42%	46.74%
W (grains WV per lb dry air)	56	35	120	775.14
acfm	30,000	29,011	1021.65	1308.73
dscfm	29,574.27	29,574.27	948	948
M (lb dry air per hr)	133,084.56	133,084.56	4266.01	4266.01
M (lb WV per hr)	1064.68	665.42	73.13	472.39
ΔQ sensible (Btu per hr)	1,215,006	735,214	64,597	131,723
ΔQ latent (Btu per hr)	1,162,977	722,535	80,669	553,896
ΔQ total (Btu per hr)	2,377,983	1,457,749	145,266	665,619

Water vapor removed (lb per hr): 399.26

These data sections represent what occurs between Point 1 and Point 2 and what happens between Point 3 and Point 4:

Process air heat removal
ΔQ sensible = 479,792 (Btu per hr)
ΔQ latent = 440,442 (Btu per hr)
ΔQ total = 920,234 (Btu per hr)

Cooling coil load
Process air = 920,234 (Btu per hr)
Heat dump back = 85,766 (Btu per hr)
Total = 1,006,000 (Btu per hr)

Regen air heat addition
ΔQ sensible = 67,126 (Btu per hr)
ΔQ latent = 453,227 (Btu per hr)
ΔQ total = 520,353 (Btu per hr)

Heater load
Regen air = 520,353 (Btu per hr)
Desiccant heat = 156,047 (Btu per hr)
Total = 676,400 (Btu per hr)

3 Liquid desiccant dehumidifier.

to a vapor as it passes through the restriction in the valve. Then the liquid, with some bubbles of flash vapor, enters the evaporator coil. The liquid refrigerant in the coil evaporates and, in doing so, absorbs heat from the air passing through the coil.

An example of a DX dehumidification system is illustrated in Fig. 1. The system includes a heat-pipe heat exchanger and an air-cooled condenser system. The heat pipe removes 538,616 Btu per hr (44.88 ton) of heat from the inlet process air and passes it to the dehumidifier process air leaving the system. The advantage is reduced cooling load and a leaving air condition that is not at or near saturation. The operating cost in Table 4 is

$20.03 per 1000 lb of water vapor removed at a process-air-leaving condition of 56.3 F and 36 grains per lb (53.9 percent RH). Dehumidification reheat systems will use the high-pressure, high- temperature gas leaving the compressor in a coil system to reheat the process air leaving the dehumidifier. Most common residential dehumidifiers use this configuration. It is important for designers to consider waste energy usage, heat wheels, or heat pipes for reheat because reheat can add a significant "added" cost to the operating cost of a DX system.

A standard DX system with no heat pipe has an operating cost of $26.37 per 1000 lb of water vapor removed—at a process-air-leaving condition of 42.7 F and 36 grains per lb (89.7 percent RH).

TABLE 4—DX with heat pipe dehumidification operating cost to remove 1000 lb of water vapor.

	Operating cost	Electric usage
1) Process air fan	$2.74 per 1000	17.36 KW per hr
2) Compressors	$16.17 per 1000	102.49 KW per hr
3) Condenser fans	$1.12 per 1000	7.11 KW per hr
4) Total	**$20.03 per l000 lb WV** @ process air discharge of 56.3 F and 36 grains WV per lb dry air	**126.96 KW/hr**

a) Electricity price: $0.06 per KWH
b) DX cooling with no energy recovery has a cost of $26.37 per 1000 lb WV with a process air discharge of 42.7 F and 36 grains WV per lb dry air.

TABLE 5—Types of dry solid adsorbents.

1) Silica gel
2) Titanium gel
3) Dry lithium chloride
4) Zeolites
5) Synthetic zeolites (molecular sieves)
6) Activated alumina
7) Synthetic polymers

Rotating dry desiccant wheel

Dry desiccants are adsorbent materials that attract moisture because of the electrical field at the desiccant surface. The field attracts water molecules that have a net opposite charge. Some of the solid adsorbents used in dry desiccant systems are illustrated in Table 5.

Sorption is the adsorption process by which a desiccant removes water vapor directly from the air. The ability of an adsorbent to attract moisture depends on the difference in vapor pressure between the desiccant surface and air. The vapor-pressure difference drives moisture from the high vapor-pressure area to the low vapor-pressure area. Dry desiccants typically have low vapor pressure at their surface and, therefore, adsorb moisture from the air. When moisture is removed from the process air stream, it produces heat of sorption (or heat of adsorption), which is composed of latent heat of condensation of the removed moisture plus additional chemical heat. The heat of sorption of the moisture removed from the air is converted to sensible heat. The amount of heat released is usually around 1080 Btu per lb WV removed to 1312 Btu per lb WV removed. The actual amount depends on the type of desiccant. The heat of sorption (sensible heat) is energy that is passed to

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TABLE 6—Dry desiccant dehumidification operating cost to remove 1000 lb of water vapor.

	Direct fired nat. gas heat	Electric usage	Indirect fired nat. gas heat	Electric reactivation heat
1) Reactivation heat	$13.67 per 1000 lb	—	$16.33 per 1000 lb	$43.19 per 1000 lb
2) Reactivation blower	$0.54 per 1000 lb	3.62 KW per hr	$0.54 per 1000 lb	$0.54 per 1000 lb
3) Process air blower	$1.25 per 1000 lb	8.37 KW per hr	$1.25 per 1000 lb	$1.25 per 1000 lb
4) Process air bypass blower	$0.35 per 1000 lb	2.34 KW per hr	$0.35 per 1000 lb	$0.35 per 1000 lb
5) Desiccant wheel-drive motor	$0.06 per 1000 lb	0.41 KW per hr	$0.06 per 1000 lb	$0.06 per 1000 lb
6) Sub-total operating cost	$15.87 per 1000 lb WV	14.74 KW per hr	$18.53 per 1000 lb WV	$45.39 per 1000 lb WV
7) Cost to cool air to 55 F	$18.37 per 1000 lb WV	122.66 KW per hr	$18.37 per 1000 lb WV	$18.37 per 1000 lb WV
8) Total operating cost	**$34.24 per 1000 lb WV** With a process air discharge of 93.6 F and 35 grains per lb	**137.40 KW per hr**	**$36.90 per 1000 lb WV**	**$63.76 per 1000 lb WV**

Notes:
a) Electricity price: $0.06 per KWH (or $17.584 per 10^6 Btu)
b) Natural gas price: $5.00 per 10^6 Btu
c) Direct fired natural gas usage: 1,095,458 Btu per hr
d) Indirect fired natural gas usage: 1,308,550 Btu per hr
e) Electric coil heat provided: 984,128 Btu per hr
f) Heat removal to cool air to 55 F: 1,247,418 Btu per hr—(103.95 ton)
g) Cooling energy requirement: 1.18 KW per ton (air cooled-rotary liquid chiller and pump)
h) Modular vertical bed system operating cost: $14.55 per 1000 lb WV @ 89.5 F and 35 grains WV per lb dry air

Heat of sorption equation (approximate)

The water vapor removed from the process air (latent heat) was adsorbed by the desiccant, which released that heat as sensible heat to the process air.

$Q_s = \text{dscfm}_p \times 1.08\ [0.625\ (W_{pi} - W_{po}]$
$= \text{dscfm}_p \times 0.675\ (W_{pi} - W_{po})$

Reactivation air heater—heat addition (approximate)

$Q_{RA} = \text{dscfm}_r \times 1.08 \times (t_{ro} - t_{ri})$

Reactivation system heat leakage passed to process air (approximate)

This is desiccant heat from the reactivation section passing heat to the process air.

$Q_{RL} = \text{dscfm}_p \times 1.08 \times k \times (t_{ro} - t_{pi})$

Process air—discharge temperature (approximate)

$t_{po} = t_{pi} + 0.750\ (W_{pi} - W_{po}) + k\ (t_{ro} - t_{pi})$

Reactivation air flow (approximate)

$\text{dscfm}_r = \text{dscfm}_p \times (t_{po} - t_{pi})/(t_{ro} - (t_{ri})$

Reactivation outlet moisture content (approximate)

$W_{ro} = W_{ri} + (\text{dscfm}_p/\text{dscfm}_r)(W_{pi} - W_{po})$

Process air through the desiccant wheel —energy loses and gains

ΔQ sensible heat gain = heat of sorption + reactivation system leakage

ΔQ latent heat loss = heat of sorption

ΔQ total = ΔQ sensible + ΔQ latent

ΔQ total = (heat of sorption + reactivation system leakage) - heat of sorption = reactivation system leakage

Note: Heat of sorption is sometimes called heat of condensation

Nomenclature

dscfm_p = process air flow through the desiccant wheel in dry standard ft^3 per min
dscfm_r = reactivation air flow through the desiccant wheel in dry standard ft^3 per min
Q_S = heat of sorption, Btu per hr
Q_{RA} = reactivation heat addition, Btu per hr
Q_{RL} = reactivation system heat leakage passed to the process air Btu per hr
t_{po} = process air temperature leaving desiccant wheel, F
t_{pi} = process air temperature entering desiccant wheel, F
W_{pi} = amount of moisture in the process air entering desiccant wheel, grains of water vaporper lb of dry air
W_{po} = amount of moisture in the process air leaving desiccant wheel, grains of water vapor per lb of dry air
W_{ri} = amount of moisture in the reactivation air (after the heater) entering the desiccant wheel, grains of water vapor per lb of dry air
W_{ro} = amount of moisture in the reactivation air leaving the desiccant wheel, grains of water vapor per lb of dry air
t_{ro} = heated reactivation air temperature entering the desiccant wheel (equals temperature of reactivation air leaving the heater)
t_{ri} = reactivation air temperature entering the heater
k = a factor that varies from 0.038 to 0.11. (Some books recommend a value of 0.07.)

continued from page 54

the process air stream, which raises the discharge air temperature of the process air stream.

As the moisture content of the desiccant rises, so does the water-vapor pressure at the desiccant surface. At some point, the vapor pressure at the desiccant surface will be the same as the air, and moisture adsorption will end. The desiccant is then taken out of the process air stream and is placed into the reactivation air stream (a scavenger air stream consisting of outside air or building air). The reactivation air stream is typically heated to a temperature of 190 to 375 F. The combination of the heat and moisture raises the vapor pressure at the desiccant surface. When the surface vapor pressure exceeds the vapor pressure of the reactivation air, moisture leaves the desiccant. This process is called reactivation.

The reactivation section, which constitutes less than half of the desiccant wheel, uses flexible seals to seal it from the adsorption or process side to minimize cross contamination. The typical leakage rate is 1 to 2 percent.

Following reactivation, the hot desiccant rotates back into the process air where the process air cools the desiccant which lowers the desiccant vapor pressure, so it can collect more moisture from the balance of the process air stream. Some of the equations used by the rotating dry desiccant manufacturers are shown in the accompanying sidebar.

A typical layout of a rotary dry desiccant system is illustrated in Fig. 2. The operating costs of the system are listed in Table 6. The operating cost for the example in Fig. 1 is $15.87 per 1000 lb of water vapor removed with a leaving air condition of 93.6 F and 35 grains per lb. If the air is too hot and has to be cooled to 55 F and 35 grains per lb, the operating cost increases to $34.24 per 1000 lb WV. Reactivation heat represents 86 percent of the operating cost for the direct-fired unit and 88 percent for the indirect-fired unit. This represents a great opportunity for the designer to use low-cost hot water from cogeneration; the use of low-cost steam; or condensate, refrigeration reject heat, or waste exhaust to preheat the reactivation air. These options could significantly reduce the operating cost. Keep in mind that in some cases it may be more economical to combine cooling and desiccant dehumidification. The technologies do complement each other since the refrigeration condenser reject heat from the cooling process can be used to preheat the reactivation air.

In process-drying applications, dry desiccant dehumidifiers are sometimes used without added cooling because the increase in temperature caused by the heat of adsorption is helpful in the drying process. However, in some applications, a provision must be made to remove the excess sensible heat from the process air after dehumidification. For this reason, Table 6 provides an added cost section for cooling the air to 55 F as an example of the possible added cost.

Liquid desiccant dehumidifier

Liquid desiccant dehumidification operates on the principal of chemical absorption of water vapor from the air. The absorbent or desiccant solution will change physically, chemically, or both during the sorption process. Some of the liquid desiccant solutions used for dehumidification are:

- Lithium chloride (LiCl)
- Lithium bromide (LiBr)
- Calcium chloride ($CaCl_2$)
- Triethylene glycol (TEG)
- Propylene glycol

Liquid absorption dehumidification is very similar to a chilled water air-washer system. When the air passes through the washer, its dew point approaches the temperature of the water supplied. Air that is more humid is dehumidified, and air that is less humid is humidified. In a similar manner, the liquid absorption dehumidifier sprays the air with a desiccant solution that has a lower vapor pressure than the vapor pressure of the entering process air stream. The liquid has a vapor pressure lower than water at the same temperature, and the air passing over the solution approaches this reduced vapor pressure. The ability to remove water vapor (or add water vapor) is determined by the temperature and concentration of the solution. The conditioner can be adjusted so that the conditioner delivers air at the desired relative humidity.

The vapor pressure of a given concentration of absorbent solution approximates the vapor-pressure values of a fixed relative humidity line on a psychrometric chart. For instance, a 40 percent concentration of lithium chloride closely approximates the 20 percent relative humidity line. Also, a 15 percent concentration is very close to the 80 percent relative humidity line. Therefore, it can be said that higher solution concentrations give lower equilibrium relative humidity and thus allow the absorbent to dry air to lower levels. Temperature also affects the absorbents' ability to remove moisture. For instance, a 25 percent solution lithium chloride has a vapor pressure of 0.37 in Hg at 70 F (same as air at 70 F and 50 percent RH). When the solution is heated to 100 F, the vapor pressure climbs to 0.99 in Hg. Therefore, the warmer the desiccant, the less moisture it can absorb. Also, if the solution vapor pressure is higher than the surrounding air, the water vapor will transfer to the air and dry the desiccant solution.

A typical system diagram is illustrated in Fig. 3. In the operation, warm, moist air is sprayed with a solution of chilled lithium chloride, which was cooled with chilled water in a plate-and-frame heat exchanger. The air is cooled and dehumidified by heat and mass transfer to the lithium chloride solution. A chiller with an air-cooled condenser section provides the chilled water to cool the lithium chloride solution. If the desired dehumidified air-moisture content is 50 grains per lb (or 48 F dew point), the water used to cool the desiccant can be 85 F cooling-tower water rather than chilled water. Consult Table 1 for additional information.

When moisture is removed from the air, the reaction liberates heat. This is the reverse of evaporation, when heat is consumed by the reaction. The heat that is generated is the latent heat of condensation of the water vapor plus the heat of solution (or the heat of mixing of the water and desic-

Table 7—Liquid desiccant dehumidification operating cost to remove 1000 lb of water vapor

	Operating cost	Electric usage
1) Hot water boiler—natural gas	$10.59 per 1000 lb	—
2) Regen air fan	0.09 per 1000 lb	0.63 KW per hr
3) Process air fan	1.67 per 1000 lb	11.14 KW per hr
4) Regen desiccant pump	0.09 per 1000 lb	0.64 KW per hr
5) Cond. desiccant pump	0.85 per 1000 lb	5.69 KW per hr
6) Hot water pump	0.27 per 1000 lb	1.83 KW per hr
7) Chilled water pump	1.02 per 1000 lb	6.82 KW per hr
8) Chiller compressors	12.33 per 1000 lb	82.07 KW per hr
9) Chiller condenser fans	1.52 per 1000 lb	10.14 KW per hr
10) Total	**$28.43 per 1000 lb WV** with a process air discharge of 55 F and 35 grains per lb	**118.96 KW per hr**

a) Electric price: $0.06 per KWH
b) Natural gas price: $5.00 per 10^6 Btu
c) Hot water boiler efficiency: 80 percent
d) Hot water boiler natural gas usage: 845,500 Btu per hr
e) Operating cost of liquid desiccant system with a natural gas engine driven chiller using jacket heat to preheat regenerator air: $19.85 per 1000 lb WV

cant). In desiccant dehumidification, this heat (approximately 1080 to 1320 Btu per lb water vapor removed) is transformed to the air, raising the air dry bulb temperature, and therefore, increasing the load on the chilled water system. The chilled water system must be sized to remove the latent heat of condensation (heat of sorption), the air sensible heat, and the residual heat load added by the regeneration process.

To remove the water extracted from the air and keep the liquid desiccant at a fixed concentration, a small percentage of the conditioner-desiccant pump flow (typically around 15 percent) is transferred to the regeneration system. The weak desiccant solution is pumped up to a heating system (plate-and-frame heat exchanger), which raises the temperature and vapor pressure of the liquid desiccant. The hot desiccant is then sprayed at a scavenger air stream (outside air or building air) with a lower vapor pressure that forces the water vapor out of the desiccant and into this air, which is exhausted outside. The dry desiccant returns to the regenerator sump. The desiccant is still a little warm, and its vapor pressure is still a little high—until it flows back to the conditioner and is cooled by the chilled water heat exchanger. Therefore, the cooling system must be sized to include this residual heat load added by the regeneration process (sometimes called heat dump back). The amount of heat and dump back is typically in the range of 50 to 350 Btu per lb water vapor removed.

Table 7 is a review of the operating costs associated with the system in Fig. 3. The total operating cost is $28.43 per 1000 lb of water vapor removed from the process air. You will note that the natural gas cost is 37 percent of the total cost. Therefore, it pays to find a source of waste heat to reduce these costs. Some possible sources are condenser-rejected heat, solar heat, or a natural gas engine-driven chiller, which produces hot water (engine heat) as well as chilled water at a reduced cost. The operating cost of a liquid desiccant dehumidification system combined with a natural gas engine-driven chiller is $19.85 per 1000 lb of water vapor removed. This combination represents a 30 percent reduction in operating energy costs.

Another option to save energy on liquid desiccant systems is to install a liquid-to-liquid-type heat exchanger (some call this an interchanger) placed between the warm desiccant leaving the regenerator and the cool desiccant entering the regenerator. By doing so, less energy is needed to regenerate the desiccant because it is warmer than when it left the regenerator. The heat exchanger will typically reduce the heat dump back to the conditioner-cooling consumption by about 65 percent and reduces the regenerator heat consumption by about 15 percent.

Conclusion

Table 8 is a summary of the operating costs for each of the large, industrial dehumidifiers using the electricity cost of $0.06 per KWH and the natural gas cost of $5.00 per 10^6 Btu. One can see that the dry desiccant systems have the lowest energy operating cost of all the systems, but they also have the highest discharge air temperatures.

If elevated process air temperature is not acceptable, the liquid desiccant or DX with heat pipe would be the choice (at 55 to 56 F discharge air temperature).

The outcome of this study will change, depending upon the actual energy costs for that area. In the United States, natural gas costs vary from $2.40 to $7.90 per 10^6 Btu and electricity goes from $0.018 to $0.15 per KWH. What this says is that every system must be evaluated based on the energy costs for that region. Also, the evaluation of operating costs should include maintenance and capital amortization.

Each system is unique, and they all offer ways to reduce energy operating costs. For example, the liquid desiccant system operating cost dropped 30 percent (from $28.43 to $19.85 per1000 lb WV removed) by incorporating a natural gas engine-driven chiller and using the hot water jacket heat to preheat the regenerator

TABLE 8—Dehumidifier operating cost comparison.

System type	Dry bulb temp	Water vapor* in the air	Operating cost (energy cost only)
1) DX with heat pipe	56.3 F	36	$20.03 per 1000 lb WV removed
2) DX without heat pipe	42.7 F	36	$26.37 per 1000 lb WV removed
3) Dry desiccant rotating wheel—direct-fired	93.6 F	35	$15.87 per 1000 lb WV removed
4) Dry desiccant rotating wheel—direct-fired with added cooling	55.0 F	35	$34.24 per 1000 lb WV removed
5) Dry desiccant vertical bed—direct-fired	89.5 F	35	$14.55 per 1000 lb WV removed
6) Liquid desiccant—with electric air-cooled chiller	55.0 F	35	$28.43 per 1000 lb WV removed
7) Liquid desiccant—with natural gas engine-driven chiller	55.0 F	35	$19.85 per 1000 lb WV removed

Notes: 1) Electricity cost: $0.06 per KWH 2) Natural gas cost: $5.00 per 10^6 Btu
*Grains of water vapor per lb of dry air

system. Also if the required dew point is 48 F (49.68 grains per lb) or above, cooling-tower water can be used in the summer. The DX systems can use heat pipes, heat wheels, and/or heat reject from the process cooling for reheat. Using purchased energy for DX reheat can raise the operating costs significantly. The major operating cost for dry desiccant systems is the reactivation heat that represents 86 percent of the total operating cost for the system in Table 6. Using alternative low-cost energy sources can reduce the operating costs significantly. DX system condenser reject heat could be used to preheat the reactivation air as well as desuperheater coil heat. For this reason, the most economical system may be a combination DX system and desiccant (solid or liquid) system. The technologies do complement each other since the refrigeration condenser reject heat from the air cooling process can be used to prevent the air entering the reactivation or regeneration section of the desiccant dehumidifier. In industrial plants, there are many sources of cheap, low-grade heat (low temperature) that can be used in the reactivation or regeneration system. It is up to the engineer involved in the equipment selection to consider these sources before a decision can be made on the type of system. **HPAC**

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Mr. Acker has 25 years of experience in industrial HVAC, hygieneology, cogeneration, and environmental engineering. He is considered an expert in the analysis of air and water vapor at atmospheric pressure or compressed. Questions or comments on this article may be directed to the author at 902-465-3548.

PART ONE

Industrial Dehumidification:

Air Flow Diagrams & Water Vapor Load Equations

Accurate water vapor load equations for industrial dehumidification systems design are difficult to find, and terminology is not standard. This article provides a thorough review of both.

By WILLIAM ACKER*,
President,
Acker & Associates,
Green Bay, Wis.

In designing dehumidification systems, one of the most important tasks is to quantify the water vapor loads that must be removed by the system. Two qualified individuals may arrive at different total moisture loads for the same building. Some of these differences occur from abbreviated or approximate equations that were developed to make the calculations easier. However, these approximate equations lose accuracy when air temperatures are higher or lower than the conditions assumed when the approximate equations were developed. This article will clarify some of these equations and present alternative equations that are more precise across a larger range of conditions.

This article will also explain the air flows used in proposals from dehumidification companies as well as the proposed flow diagrams from dehumidification equipment suppliers. I have discovered a great amount of uncertainty over the air flows in these diagrams.

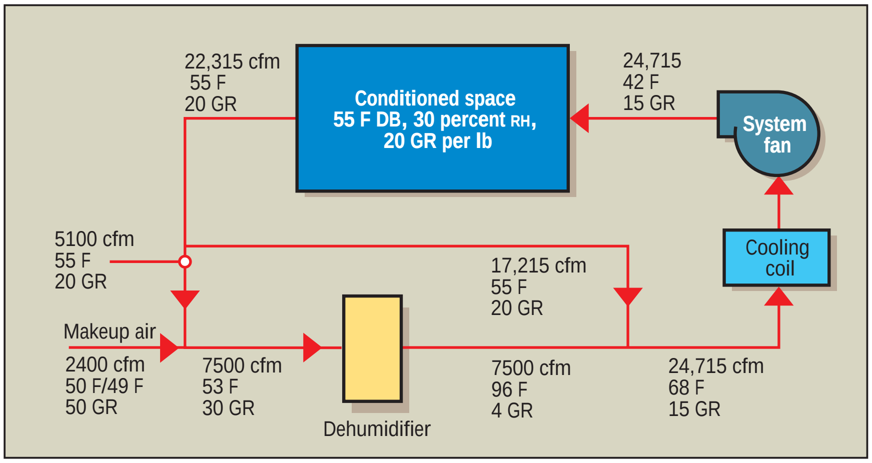

FIGURE 1 Dry desiccant systems flow diagram.

DEHUMIDIFICATION INDUSTRY EQUATIONS AND TERMS

Dehumidification manufacturers like to develop air flow diagrams of

**William Acker is a member of HPAC's Editorial Advisory Board.*

their entire systems. Mass flow analysis is used in these diagrams because mass flow does not change if there is a temperature or pressure change. Mass flows can also be added and subtracted. The actual air flows or acfm (cu ft per min) will change if the temperature or pressure changes. Also, acfm values cannot be added or subtracted because they are at different air densities. The dehumidification industry chose a type of mass flow analysis that is flow in dry standard cu ft per min (dscfm) at a common air density of 0.075 lb dry air per dry standard cu ft. The acfm flows are converted to these flows for illustration in the air system diagrams (shown in Fig. 1 and Fig. 2).

As mentioned earlier, dscfm air flow is a flow at a common air density of 0.075 lb dry air per dry standard cu ft. I

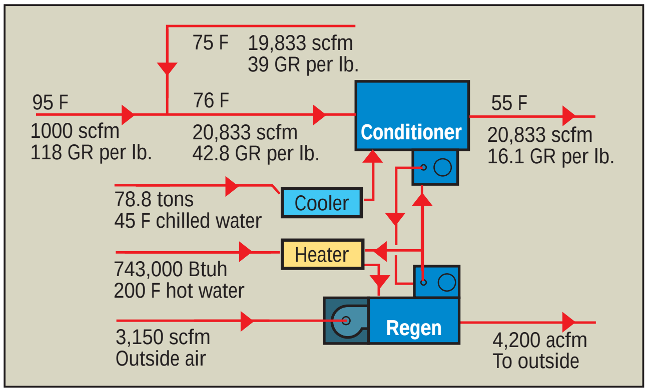

FIGURE 2 Liquid desiccant system flow diagram.

prefer to use the term dscfm to avoid any possible confusion with acfm or the fan industry term scfm. In Fig. 1, one manufacturer uses cfm to represent the dry standard air flow. When you look at this diagram, it is easy to tell that the flows are dry standard air flows and not acfm. If you look at the flow in and out of the dehumidifier, the flows are both 7500 cfm. This is not possible with acfm because this flow is made of both air and water vapor; therefore, there is a loss (of water vapor or cfm) as it travels through the dehumidifier. Listed below is an example of an acfm flow broken down into dry air flow and water vapor flow:

Air pressure = 29.921 in. Hg
Air temperature = 70 F
Relative humidity = 100 percent
Total air flow = 100,000 acfm
Dry air flow = 97,530 cfm
Water vapor flow = 2470 cfm

Also, note that there is a change in temperature across the dehumidifier that would also cause a change in the acfm flow.

Fig. 2 is a diagram prepared by another dehumidifier manufacturer. The air flows are in dscfm, but in this case, they use the term scfm. This scfm should not be confused with the fan industry scfm, which is a type of mass flow that represents the combined air and water vapor flow. By looking at Fig. 2, you can see that the flow into the dehumidifier (or conditioner) 20,833 scfm is the same as the flow leaving; therefore, this is a dry air mass flow. If the entering flow is fan industry 20,833 scfm, the leaving flow would have dropped to 20,754 scfm due to the removal of water vapor (79 scfm or 355.41 lb of water vapor per hr). The next section on fan industry scfm will help you to understand the difference between fan and dehumidification scfm terms.

The equations used by the dehumidification industry to convert the actual air flows to dry standard air flows are listed below:

$$\text{dscfm}\frac{(\text{dry std cu ft})}{\text{min}} = \frac{\text{dry air mass flow}\dfrac{(\text{lb dry air})}{\text{min}}}{0.075\dfrac{\text{lb dry air}}{\text{dry std cu ft}}} \qquad (1)$$

$$= \frac{\text{acfm}\dfrac{(\text{cu ft wet air})}{\text{min}}}{0.075\dfrac{\text{lb dry air}}{\text{dry std cu ft}} \times v\dfrac{(\text{cu ft wet air})}{\text{lb dry air}}} \qquad (2)$$

$$= \frac{\text{acfm}\dfrac{(\text{cu ft wet air})}{\text{min}} \times 13.3333\dfrac{\text{dry std cu ft}}{\text{lb dry air}}}{v\dfrac{(\text{cu ft wet air})}{\text{lb dry air}}} \qquad (3)$$

Where:

$\text{dscfm}\frac{(\text{dry std cu ft})}{\text{min}}$ = This is the dry standard air flow used by the dehumidification industry.

$\text{acfm}\frac{(\text{cu ft wet air})}{\text{min}}$ = This is the actual air flow of dry air and water vapor.

$v\frac{(\text{cu ft wet air})}{\text{lb dry air}}$ = This is the air-specific volume. The value can be found in psychrometric charts provided that the barometric pressure of the chart is the same as the total pressure (barometric plus static pressures) of the acfm flow.

$0.075\frac{\text{lb dry air}}{\text{dry std cu ft}}$ = This is the standard air density for the dehumidification industry. The inverse of this value is 13.3333.

$\text{Dry air mass flow}\frac{(\text{lb dry air})}{\text{min}}$ = This is the total amount of dry air mass flow.

FAN INDUSTRY EQUATIONS AND TERMS

Fans must be selected based on the actual flow rate and the actual density at the inlet to the fan. Some fan manufacturers prefer to specify the flow rate based on standard inlet conditions. Fan performance curves are developed (from a series of laboratory tests) at these standard conditions. Listed below are the equations that allow designers to convert from the actual conditions to the standard conditions:

Where:

$$\text{acfm (cu ft per min)} = \frac{\text{total mass flow (lb per min)}}{\text{density of moist gas (lb total per cu ft)}} \quad (4)$$

$$\text{scfm (std cu ft per min)} = \frac{\text{total mass flow (lb per min)}}{\text{std density 0.075 (lb total per std cu ft)}} \quad (5)$$

$$\text{acfm (cu ft per min)} = \text{scfm (std cu ft per min)} \times \frac{\text{0.075 (lb total per std cu ft)}}{\text{density of moist gas (lb total per cu ft)}} \quad (6)$$

$$\begin{array}{l}\text{Density of moist gas} \\ \text{(lb total per cu ft)}\end{array} = \frac{\text{MW wet mix (lb}_m\text{ per mole)} \times \text{P (lbf per sq ft)}}{1545.43 - \text{lb}_f\text{ per mole} \times \text{T(°R)}} \quad (7)$$

acfm (cu ft per min) =

Actual cubic feet per minute. It represents the volume of dry gas and water vapor flowing at a specified point in a system. In fan sizing, this would be the flow entering the fan.

Density of moist gas (lb total per cu ft) =

The ratio of the mass of a substance to its volume. The fan gas density or fan air density is the total density at the fan inlet.

Standard density (lb total per std cu ft) =

Some fan manufacturers like to develop fan performance curves based on a standard gas or air density of 0.075 lb total per std cu ft. The Air Movement and Control Association (AMCA) indicates that this density is substantially equivalent to air at a temperature of 68 F, 50 percent RH, and a pressure of 29.92 in. of mercury.

scfm (std cu ft per min) =

The standard gas or air flow rate entering the fan at an inlet density of 0.075 lb total per std cu ft.

MW wet mix (lbm per mole) =

The molecular weight of the dry gas (or dry air) and water vapor mixture.

P (lbf per sq ft) =

The total pressure at the inlet to the fan. The pressure represents the barometric pressure plus the static pressure (or gauge pressure).

1545.43 ft - lbf per mole - ˚R =

Universal gas constant.

T(˚R) =

Absolute gas or air temperature at the inlet to the fan in degrees Rankine (˚ R = ˚ F + 459.67).

DSCFM FLOW VERSUS SCFM FLOW

The following examples show how the dehumidification industry dscfm flow compares to the fan industry scfm flow.

Example 1:

Air pressure = 29.921 in. Hg
Air temperature = 70 F
Relative humidity = 100 percent
Humidity ratio = 109.93 grains WV per lb dry air
Moist air density = 0.074190 lb wet air per cu ft wet air
Specific volume = 13.6906 cu ft wet air per lb dry air
acfm flow = 40,000 cu ft wet air per min
Dehumidification industry dscfm = 38,956 dry std cu ft per min
Fan industry scfm = 39,568 std cu ft per min

Example 2:

Air pressure = 29.921 in. Hg
Air temperature = 200 F
Relative humidity = 50 percent
Humidity ratio = 2809 grains WV per lb dry air
Moist air density = 0.051216 lb wet air per cu ft wet air
Specific volume = 27.3599 cu ft wet air per lb dry air
acfm flow = 40,000 cu ft wet air per min
Dehumidification industry dscfm = 19,493 dry std cu ft per min
Fan industry scfm = 27,315 std cu ft per min

Example 1 shows that at 70 F, the dscfm flow is very close to the fan industry scfm flow (only 1.6 percent variation). However, at more elevated temperatures, such as in Example 2, the fan industry scfm flow is 40 percent higher than the dscfm flow.

WATER VAPOR LOADS DUE TO AIR FLOW

Water vapor loads on industrial buildings come from many sources. Listed below are a few of these sources of water vapor:

- ❒ People
- ❒ Permeation through walls, roofs, and floors
- ❒ Moisture from products and packaging materials
- ❒ Evaporation from open tanks or wet surfaces
- ❒ Product dryer leakage
- ❒ Open combustion
- ❒ From air flow
  - Air leakage through cracks and holes
  - Air leakage through conveyor openings
  - Intermittent door openings
  - Building-to-building air infiltration
  - Makeup air

In many cases, water vapor loads by air flow are a major contributor to the total building vapor load. In the research work for this article, I came across a number of approximate equations that are used to calculate the water vapor load from air flow. Approximate equations can be fairly accurate as long as the air conditions are close to 70 F air temperature.

For more information on water vapor permeation loads, consult the June 1998 issue of *HPAC* magazine. The next few sections will compare approximate and exact equations for selected water vapor sources. **HPAC**

This article will continue in HPAC*'s July issue. Part II will cover moisture from air leakage, water vapor load from makeup air, and water vapor removed by dehumidifiers.*

Industrial Dehumidification:

Air Flow Diagrams & Water Vapor Load Equations

PART TWO

Accurate water vapor load equations for industrial dehumidification systems design are difficult to find, and terminology is not standard. This article provides a thorough review of both.

By WILLIAM ACKER*, President,
Acker & Associates, Green Bay, Wis.

This is the second article of a two-part series on industrial dehumidification. The first article in the series, which covered air flow, can be read in HPAC Engineering's May 1999 issue.

WATER VAPOR LOADS DUE TO AIR FLOW

Water vapor loads on industrial buildings come from many sources. Listed below are a few of these sources of water vapor:

- ❒ People
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- ❒ Moisture from products and packaging materials
- ❒ Evaporation from open tanks or wet surfaces
- ❒ Product dryer leakage
- ❒ Open combustion
- ❒ Air flow
  - Air leakage through cracks and holes
  - Air leakage through conveyor openings
  - Intermittent door openings
  - Building-to-building air infiltration
  - Makeup air

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For more information on water vapor permeation loads, consult the June 1998 issue of HPAC Engineering. The next few sections will compare approximate and exact equations for selected water vapor sources.

MOISTURE FROM AIR LEAKAGE

Equations (8) to (12) can be used to calculate moisture for air leakage through cracks, holes, and conveyor openings. Equations (8) to (11) were taken from engineering books or from manuals prepared by dehumidification companies. Notice that the engineering units do not properly cancel out, which is why they are considered approximate equations.

$$M\frac{\text{(grains WV)}}{\text{hr}} = \text{Velocity}\frac{\text{ft}}{\text{min}}\times 60\frac{\text{min}}{\text{hr}}\times \text{Area (sq ft)} \times d\frac{\text{(lb wet air)}}{\text{(cu ft wet air)}}\times (W_o - W_i)\frac{\text{grains WV}}{\text{lb dry air}} \quad (8)$$

$$= \text{acfm}\frac{\text{(cu ft wet air)}}{\text{min}}\times 60\frac{\text{min}}{\text{hr}}\times d\frac{\text{(lb wet air)}}{\text{(cu ft wet air)}} \times (W_o - W_i)\frac{\text{grains WV}}{\text{lb dry air}} \quad (9)$$

$$= \text{acfm}\frac{\text{(cu ft wet air)}}{\text{min}}\times 60\frac{\text{min}}{\text{hr}} \times 0.075\frac{\text{(lb wet air)}}{\text{(cu ft wet air)}}\times (W_o - W_i)\frac{\text{grains WV}}{\text{lb dry air}} \quad (10)$$

$$M\frac{\text{(grains WV)}}{\text{hr}} = \text{acfm}\frac{\text{(cu ft wet air)}}{\text{min}}\times 4.5\times (W_o - W_i)\frac{\text{grains WV}}{\text{lb dry air}} \quad (11)$$

Where:

W_o = The higher moisture level of the air outside the

*William Acker is a member of HPAC Engineering's Editorial Advisory Board.

room that is entering with the air flow. It is an absolute humidity term. (ASHRAE calls this the Humidity Ratio.)

Wi = Moisture level of the air inside the room which in this case is at a lower absolute humidity. (ASHRAE calls this the Humidity Ratio.)

$$d\frac{(\text{lb wet air})}{(\text{cu ft wet air})} = \text{Is the density of moist air entering the room}$$

Equation (12) can be used for the same calculations. It is an exact equation because the engineering units convert to grains of water vapor per hr.

$$M\frac{(\text{grains WV})}{\text{hr}} = \text{acfm}\frac{(\text{cu ft wet air})}{\text{min}} \times 60\frac{\text{min}}{\text{hr}} \times \frac{1}{v\dfrac{(\text{cu ft wet air})}{(\text{lb dry air})}} \times (W_o - W_i)\frac{\text{grains WV}}{\text{lb dry air}} \quad (12)$$

$v\frac{(\text{cu ft wet air})}{(\text{lb dry air})}$ = Specific volume of the air flowing into the room. This value can be found in psychrometric charts or calculated. When using the charts, the combined air barometric and static pressure should equal the total air pressure of the chart.

Equations (11) and (12) are applied and compared in the example below to show how results from approximate and exact equations can vary under different conditions. Equation (11), which is approximate, calculates a water vapor load 20.18 percent over the exact equation (12). Equation (11) is more accurate if the entering air flow is close to 70 F. Engineers preferring the exact equation will need a psychrometric chart to obtain the entering air specific volume, or a psychrometric computer program that can calculate the air mixture properties.

Example conditions:

Room conditions

Air pressure: 29.921 in. Hg
Dry bulb temp: 70 F
Moisture level: 35 grains WV/lb dry air
Relative humidity: 32.38 percent

Entering air flow conditions

Air pressure: 29.921 in. Hg
Dry bulb temp: 120 F
Moisture level: 420 grains WV/lb dry air
Relative humidity: 76.44 percent
Specific volume: 16.02379 cu ft wet air/lb dry air
Air flow acfm: 200 cu ft per min

Example calculations:

$$200\frac{\text{cu ft wet air}}{\text{min}} \times 4.5 \times (420 - 35)\frac{\text{grains WV}}{\text{lb dry air}} = 346{,}500\frac{\text{grains WV}}{\text{hr}} \quad (11)$$

$$200\frac{\text{cu ft wet air}}{\text{min}} \times 60\frac{\text{min}}{\text{hr}} \times \frac{\text{lb dry air}}{16.02379\text{ cu ft wet air}} \times (420 - 35)\frac{\text{grains WV}}{\text{lb dry air}} = 288{,}321\frac{\text{grains WV}}{\text{hr}} \quad (12)$$

WATER VAPOR LOAD FROM MAKEUP AIR

Equation (12) can also be used to calculate the water vapor load from makeup air. Let's compare it to approximate Equations (13) and (14). In this example, equations (13) and (14) produced water vapor loads that were 7.27 and 12.63 percent above the exact equation (12) water vapor load. The error is a direct result of the assumed entering air specific volume. Note that the makeup air specific volume will vary with the entering air psychrometric properties. Therefore, you cannot select a standard value for specific volume and expect the equation to be exact. For this reason, equations (13) and (14) are approximate equations. Equations (13) and (14), which can be found in many engineering books and dehumidification manuals, will be fairly accurate as long as the entering makeup air is close to the selected specific air volume.

$$M\frac{(\text{grains WV})}{\text{hr}} = \text{acfm}\frac{(\text{cu ft wet air})}{\text{min}} \times 60\frac{\text{min}}{\text{hr}} \times \frac{\text{lb dry air}}{14\text{ cu ft wet air}} \times (W_o - W_i)\frac{\text{grains WV}}{\text{lb dry air}} \quad (13)$$

$$M\frac{(\text{grains WV})}{\text{hr}} = \text{acfm}\frac{(\text{cu ft wet air})}{\text{min}} \times 60\frac{\text{min}}{\text{hr}} \times 0.075\frac{\text{lb}}{\text{cu ft}} \times (W_o - W_i)\frac{\text{grains WV}}{\text{lb dry air}} \quad (14)$$

$$M\frac{(\text{grains WV})}{\text{hr}} = \text{acfm}\frac{(\text{cu ft wet air})}{\text{min}} \times 60\frac{\text{min}}{\text{hr}} \times \frac{1}{v\dfrac{(\text{cu ft wet air})}{(\text{lb dry air})}} \times (W_o - W_i)\frac{\text{grains WV}}{\text{lb dry air}} \quad (12)$$

Example conditions:

Inside room conditions

Air pressure: 29.921 in. Hg
Dry bulb temperature: 70 F
Moisture level: 35 grains WV/lb dry air
Relative humidity: 32.38 percent

Entering makeup air

Air pressure: 29.921 in. Hg
Dry bulb temperature: 100 F
Moisture level: 280 grains WV/lb dry air
Relative humidity: 93.67 percent
Specific volume: 15.01722 cu ft wet air/lb dry air
Air flow acfm: 2000 cu ft per min

Example calculations:

$$2000\text{ cfm} \times 60 \times \frac{1}{14} \times (280 - 35) = 2{,}100{,}000\text{ grains WV per hr} \quad (13)$$

$$2000\text{ cfm} \times 60 \times 0.075 \times (280 - 35) = 2{,}205{,}000\text{ grains WV per hr} \quad (14)$$

$$2000\text{ cfm} \times 60 \times \frac{1}{15.01722} \times (280 - 35) = 1{,}957{,}753\text{ grains WV per hr} \quad (12)$$

WATER VAPOR REMOVED BY DEHUMIDIFIERS

Dehumidification systems remove water vapor from the process air that travels through the unit. This section looks at the equations used to determine the humidity ratio of the process air entering and leaving the unit as well as equations used to estimate the amount of water vapor removed when the inlet and discharge humidity ratios and air flow are known. Note that some of the equations are the same as the equations used in preceding sections. The equations listed as

TABLE 1

	Into dehumidifier	Dehumidifier discharge
Air pressure	29.921 in. Hg	29.921 in. Hg
Air temperature	200 F	
Relative humidity	50 percent	
Humidity ratio	280 grains WV per lb dry air	35 grains WV per lb dry air
Moist air density	0.051216 lb wet air per cu ft wet air	
Specific volume	27.3599 cu ft wet air per lb dry air	
acfm flow	40,000 cu ft wet air per min	
Dry air mass flow	87,719.78 lb dry air per hr	87,719.78 lb dry air per hr
Water vapor flow	35,197.64 lb water vapor per hr	
Fan industry scfm	27,315 std cu ft per min	
Dehumidification industry dscfm	19,493 dry std cu ft per min	19,493 dry std cu ft per min

approximate can be very accurate if the process air flow temperature is close to 70 F (Table 1).

In this series of equations and calculations, approximate equations (15) and (11) produced water vapor removals that were 40 and 105 percent above the exact equations (16) and (17). The approximate equations can be fairly accurate if the air entering the dehumidifier is close to 70 F. Engineers that desire greater accuracy can use the psychrometric chart to get the specific volume needed to make the conversion from acfm to dscfm or purchase psychrometric programs that can calculate the value for them.

$$
\begin{aligned}
M\frac{(\text{grains WV})}{\text{hr}} &= \text{scfm}\frac{(\text{std cu ft air})}{\text{min}} \times 60\,\frac{\text{min}}{\text{hr}} \\
&\times 0.075\,\frac{\text{lb wet air}}{\text{std cu ft air}} \times (W_o - W_i)\,\frac{\text{grains WV}}{\text{lb dry air}} \qquad (15) \\
&= 27{,}314.98 \times 60 \times 0.075 \times (2809 - 35) \\
&= 340{,}972{,}895
\end{aligned}
$$

Note: This equation uses the fan industry scfm flow entering the dehumidifier. The engineering units do not properly cancel out on the right side of the equation. This is why the equation accuracy is listed as approximate.

$$
\begin{aligned}
M\frac{(\text{grains WV})}{\text{hr}} &= \text{acfm}\frac{(\text{cu ft wet air})}{\text{min}} \times 60\,\frac{\text{min}}{\text{hr}} \\
&\times 0.075\,\frac{\text{lb wet air}}{\text{cu ft std air}} \times (W_o - W_i)\,\frac{\text{grains WV}}{\text{lb dry air}} \qquad (10)
\end{aligned}
$$

$$
\begin{aligned}
M\frac{(\text{grains WV})}{\text{hr}} &= \text{acfm}\frac{(\text{cu ft wet air})}{\text{min}} \times 4.5 \\
&\times (W_o - W_i)\,\frac{\text{grains WV}}{\text{lb dry air}} \qquad (11) \\
&= 40{,}000 \times 4.5 \times (2809 - 35) \\
&= 499{,}320{,}000
\end{aligned}
$$

Note: This equation appears in some dehumidification publications. Because the engineering units do not properly cancel out on the right side of the equation, it is considered an approximate equation.

$$
\begin{aligned}
M\frac{(\text{grains WV})}{\text{hr}} &= \text{dscfm}\frac{(\text{dry std cu ft})}{\text{min}} \times 60\,\frac{\text{min}}{\text{hr}} \\
&\times 0.075\,\frac{\text{lb dry air}}{\text{dry std cu ft}} \times (W_o - W_i)\,\frac{\text{grains WV}}{\text{lb dry air}} \qquad (16) \\
&= 19{,}493 \times 60 \times 0.075 \times (2809 - 35) \\
&= 243{,}334{,}670
\end{aligned}
$$

Note: Uses the dehumidification industry dscfm flow entering the dehumidifier.

$$
\begin{aligned}
M\frac{(\text{grains WV})}{\text{hr}} &= M\frac{(\text{lb dry air})}{\text{hr}} \times (W_o - W_i)\,\frac{\text{grains WV}}{\text{lb dry air}} \qquad (17) \\
&= 87{,}719.78\,\frac{\text{lb dry air}}{\text{hr}} \times (2809 - 35) \\
&= 243{,}334{,}670
\end{aligned}
$$

Note: This equation can be found in examples of the ASHRAE Handbook of Fundamentals; *in* Modern Heating and Ventilating Systems Design, *by George E. Clifford; and in* Fan Engineering, *by Buffalo Forge.*

CONCLUSION

Many of the flow diagrams presented in articles, books, engineering manuals, and proposals from dehumidification companies do not indicate the engineering units for the flows in the diagrams. As indicated in this article, the flows are illustrated in cfm or scfm with no explanation. In most cases, the flows are in dry standard cubic feet per minute.

Over the years, I have been contacted by many engineers over the issue of calculated water vapor load variances from different equations. In most cases, the approximate equations are equations that have been shortened to make the calculations easier for engineers. If you have any questions on your equations, check the engineering units to make sure that they properly cancel out to grains of water vapor per hr, or lb of water vapor per hr. **HPAC**

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