

AIR-PROCESSING EQUIPMENT

Cooling, heating, humidifying, dehumidifying, and cleaning of air are some of the processes for which mechanical equipment is needed in HVAC systems. These kinds of air-processing devices are examined in this material, along with air-to-air energy recovery equipment. For additional information on these topics, the reader is referred to Chapters 21 through 29 and 41 of the 2020 *ASHRAE Handbook—HVAC Systems and Equipment*. Chapters 3 and 4 of the 2021 *ASHRAE Handbook—Fundamentals* provide the heat transfer background for much of this equipment.

1 Air-Handling Equipment

Air-handling units consist of the equipment that filter, heat, humidify, cool, dehumidify, and move the air. The three major types of air handlers are

- Factory-fabricated units
- Built-up (field-erected) units
- Customized units

Each category contains both single-zone and multizone units.

Factory-fabricated units usually contain the fan and cooling and/or heating coils and filter assembled in a cabinet, usually of sheet metal. Mixing boxes are also included as desired. Standard arrangements and options available with factory-assembled units include

- Draw-through or blow-through
- Horizontal or vertical assembly
- Chilled water or refrigerant cooling coils
- Hot water, steam, or electric heating coils
- Forward-curved, backward-curved, airfoil, axial, or plug fans
- Flat or “V” bank filter sections
- Mixing boxes with damper assemblies

Factory-fabricated units are usually less expensive than field-erected built-up systems and the delivery time is usually shorter than for the purchase and assembly of a built-up unit.

Rooftop units may be considered a special type of factory-fabricated unit, being similar to such indoor units, except that they are located on the roof, saving valuable space that would otherwise be needed for the mechanical equipment room. Packaged rooftop units are sometimes completely self-contained including the condensing unit and usually either a gas-fired or electric heating section. On the negative side, rooftop units generally require higher maintenance and have a shorter service life than units protected in mechanical equipment rooms.

Built-up air-handling units consist of separate casings enclosing fans, coils, filters, mixing boxes, and pleums. Built-up unit casings may be factory made or fabricated by a local sheet metal contractor. Dimensions of built-up units can often be varied for the floor area available, being limited only by the coil and fan selections. There is almost no limit to the capacity of built-up units.

Customized air-handling units are normally selected to provide either a higher level of quality or special component configurations for a special application.

2 Cooling Coils

Ceiling coils are used to cool an airstream under forced convection. Such equipment may consist of a single coil section or several individual coil sections built into banks. Coils are also used extensively as compo-

nents in central station air-handling units, room terminals, and in factory-assembled, self-contained air conditioners.

Chilled water, brines, or volatile refrigerants are the usual cooling media. Cooling coils that use relatively high temperature water usually do not dehumidify the air; however, most coil equipment is designed to remove sensible heat and dehumidify simultaneously. The coil assembly should include a means to protect the coil from dirt accumulation and to keep dust and foreign matter out of the conditioned space.

2.1 Coil Construction and Arrangement

Coils include bare tubes or pipe, and those with extended or finned surfaces. The design of coils with extended surfaces on the air side considers the materials, fin size and spacing, ratio of extended surface area to that of the tube area, tube nesting center dimensions, staggered or in-line tube arrangement, and use of turbulators.

Staggered tubes increase the total heat transfer over that of the in-line arrangement, and turbulators may also be used to enhance total heat transfer efficiency. The surface arrangement has a great effect on the air-film heat transfer resistance and associated air-side pressure drop. Several arrangements are illustrated in Figure 1.

In fin or extended surface coils, the external surface of the tubes is **primary**, and the fin surface is **secondary**. The primary surface consists of rows of round tubes or pipes, which may be staggered or placed in line with respect to the airflow. Some tubes are flattened or have nonround internal passageways. The inside surface of the tubes is usually smooth and plain, but some designs have internal fins or turbulence promoters (either fabricated or extruded) to provide turbulence and additional inside surface area for enhancing performance. Individual tubes are generally interconnected by return bends, sometimes together with hairpin tubes to form the required serpentine arrangement of multipass tube circuits. Some flattened tubes are folded into continuous serpentine circuits with fins metallurgically bonded between adjacent tube passes. Numerous fin arrangements are used; the most common are smooth spiral, crimped spiral, flat plate, and configured plate. A good thermal bond between the tube and the fin must be maintained permanently to ensure low resistance to heat transfer from fin to tube.

In some coils, fins are wound under pressure onto the tubes in order to upset the metal slightly at the fin root. They are then coated with solder while the fin and tube are still revolving to ensure a uniform solder coating. In other types of coils, the spiral fin may be knurled into a shallow groove on the exterior of the tube. The tube may be expanded after the fins are assembled, or the tube-hole flanges of a flat or configured fin may be made to override those in the preceding fin and so compress them on the tube. Some construction techniques even form the fin from the material of the tube itself.

Cooling coils for water or for volatile refrigerants frequently have aluminum fins and copper tubes, although copper fins on copper tubes are also used and the combination of aluminum fins on aluminum tubes is finding use. Adhesive bonding is sometimes used in making header connections, return bends, and fin-tube joints, particularly for aluminum-to-aluminum joints. Many types of lightweight, extended-surface cooling coils are made for both heating and cooling. Tube outside diameters are commonly 1/4, 3/16, 3/8, 5/8, 3/4,

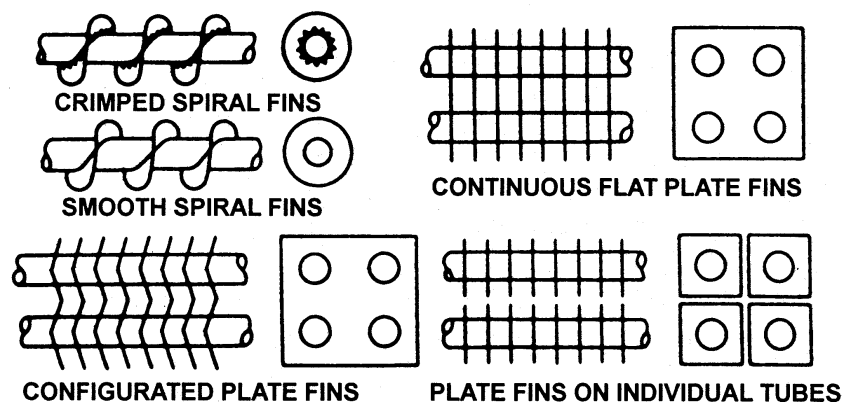


Fig. 1 Types of Fin-Coil Arrangements

and 1 in. (6.4, 9.5, 12.7, 15.9, 19.1, and 25 mm), and fins are spaced from 3 to 14 per inch (1.8 to 8.5 mm apart). The tube spacing varies from about 5/8 to 2 1/2 in. (16 to 64 mm) on centers, depending on the width of individual fins and other performance considerations. Fin spacing should be chosen for the duty to be performed, with special attention paid to air friction, prevention of water carryover, possible lint accumulation, and, especially at lower temperatures, frost accumulation.

2.2 Water Coils

Water coil performance depends on eliminating air from the water circuit and properly distributing the water. Unless the system is vented, air may accumulate in the coil circuits, which reduces thermal performance and may cause noise or vibration. Air vent connections are usually provided on the coil water headers. Depending on performance requirements, the water velocity inside tubes usually ranges from approximately 1 to 8 ft/s (0.3 to 2.4 m/s), and the design water pressure drop across coils varies from about 5 to 50 ft of water (15 to 150 kPa).

A variety of circuit arrangements, and combinations thereof, for varying the number of parallel water flow passes within the tube core are usually available. Some typical arrangements are shown in Figure 2.

2.3 Direct-Expansion Refrigerant Coils

Direct-expansion coils present more complex problems of cooling fluid distribution than water or brine coils. The coil should be effectively and uniformly cooled throughout, and the compressor must be protected from entrained, unevaporated refrigerant. Direct-expansion coils are used on both flooded and dry-expansion refrigeration systems.

The **flooded system** is mainly used for low-temperature applications where a small temperature difference between the air and refrigerant is advantageous. However, a relatively large volume of refrigerant is required, together with extra components such as a surge tank and interconnecting piping. Other applications of direct-expansion coils generally use dry expansion. For **dry-expansion systems**, the most commonly used refrigerant liquid metering devices are the capillary tube and the thermostatic expansion valve.

The **capillary tube system** is applied on evaporator coils in factory-assembled, self-contained air conditioners up to approximately 10 tons (35 kW) capacity and is used extensively on the smaller capacity models such as window or room units. In this system, the bore and length of a capillary tube are sized so that at full

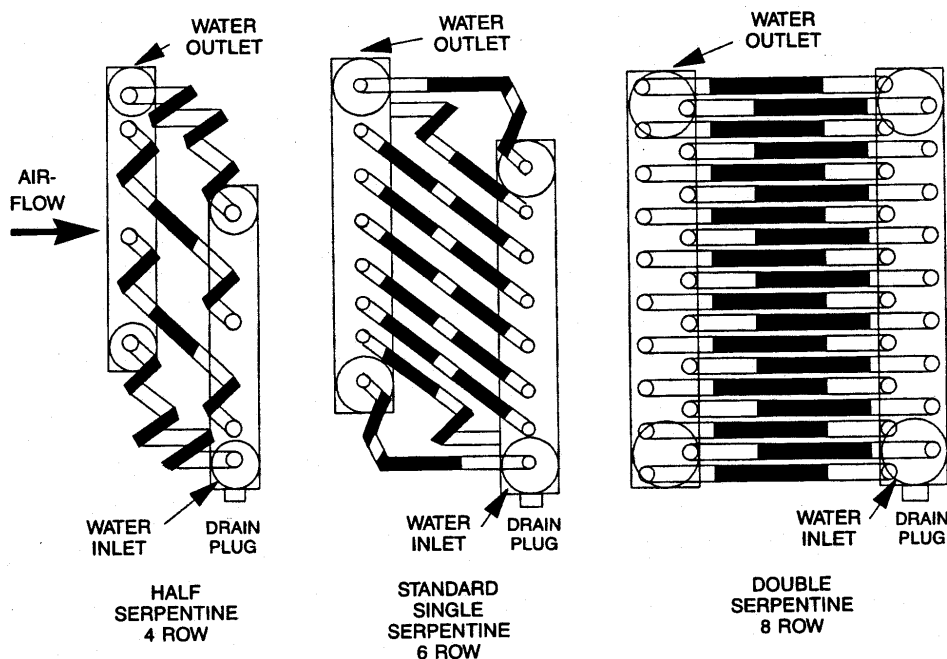


Fig. 2 Typical Water Circuit Arrangements

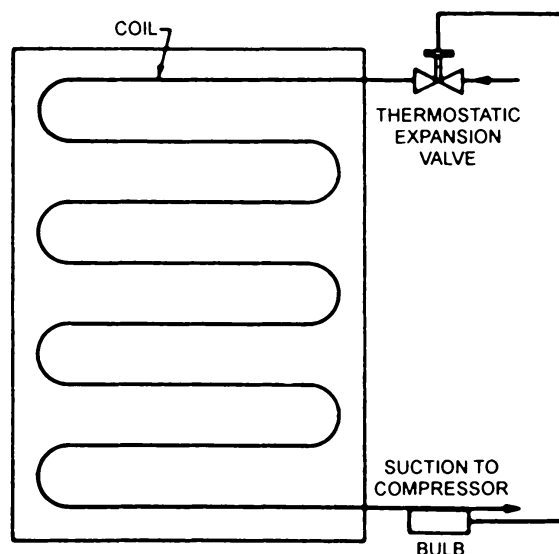


Fig. 3 Dry Expansion Coil with Thermostatic Expansion Valve

load, just enough refrigerant liquid is metered from the condenser to the evaporator coil to be completely evaporated.

The **thermostatic expansion valve system** is common for all dry-expansion coil applications, particularly for field-assembled coil sections in central air-handling units, as well as for the larger factory-assembled hermetic air conditioners. A schematic typical of a coil and thermostatic expansion valve assembly is shown in Figure 3. The thermostatic expansion valve automatically regulates the rate of refrigerant liquid flow to the coil in direct proportion to the rate of evaporation of refrigerant liquid in the coil, thereby maintaining the superheat at the coil suction outlet within the usual predetermined limits of 6 to 10°F (3 to 6°C).

2.4 Coil Selection

When selecting a coil, the following factors must be considered:

- Job requirements: cooling, dehumidifying, and the capacity required to maintain balance with other system components, for example, compressor equipment in the case of direct-expansion coils.
- Temperature of entering air: dry bulb only if there is no dehumidification; dry and wet bulb if moisture is to be removed.
- Temperature of entering, chilled water or evaporating pressure of refrigerant and type of refrigerant.
- Temperature of leaving air dry bulb and wet bulb or dew point.
- Available cooling media and its operating temperature and quantity.
- Space and dimensional limitations.
- Air quantity limitations.
- Allowable frictional resistances in cooling media piping system (including coils).
- Allowable frictional resistances in air circuit (including coils).
- Characteristics of individual coil designs.
- Individual installation requirements, such as the type of automatic control to be used or the presence of a corrosive atmosphere.
- Coil air face velocity.
- To prevent water carryover from dehumidifying coils, face velocities should not exceed 500 fpm (2.54 m/s), and fin density should be no greater than 8 fins per inch (fins 3 mm apart).

Coil ratings and selection procedures are usually presented in one of two ways:

Basic Data Method. Coil performance parameters are published in the form of tables or charts from which the coil row depth is calculated after determining the required coil sensible and total heat capacities and other design variables from the job conditions. The initial selection generally indicates a nonintegral row

depth requirement. It is frequently necessary to recheck and reselect the coil to match more closely the required air and cooling fluid conditions with the integral row depth actually installed, particularly for dehumidifying coils. The method is generally used in selecting coils or coil banks for field assembly, since there is a vast number of size and row depth combinations available.

Unit Rating Method. Performance for specific combinations of coil face area and row depth are presented in tables or charts. This method provides a direct selection of specific coils to match the required capacity under the job conditions. This method is frequently used in selecting coils for central station air-handling units and also in determining performance for factory-assembled, self-contained air conditioners.

With either method, various combinations of coil face area, row depth, air velocity, and air quantity may be chosen to do the same job. Coil selection requires understanding each case, and selection should be based on an economic analysis of the plant as a whole.

Most coil manufacturers and some commercial software providers have programs to assist in the selection and diagnosis of cooling coils.

Table 1 provides an illustration of coil performance data as may be found in a typical cooling coil catalog.

Coils that operate wet, particularly those with enhanced (configurated) fins tend to build up dirt on the fin and tube surfaces so they must be capable of being, first, visually inspected for fouling and, second, cleaned. Both of these are difficult without sufficient space on each side of the coil. Fin spacing closer than 1/8 of an inch or enhanced fins are not recommended.

Improper selection of the cooling coil is the most common cause of performance failures in air-conditioning systems.

2.5 Application Range

Based on information in AHRI Standard 410, dry surface (sensible cooling) coils and dehumidifying coils (which both cool and dehumidify), particularly for field-assembled coil banks, factory-assembled coil banks, or factory-assembled central station air conditioners using different combinations of coils, are usually rated within the following limits:

Entering air temperature	65°F to 100°F (18°C to 38°C)
Entering air wet bulb	60°F to 85°F (15°C to 30°C)
Air face velocity	300 to 800 fpm (1.5 to 4.0 m/s) (sometimes as low as 200 and as high as 1500 fpm)
Refrigerant saturated	30°F to 55°F (−1°C to 13°C) at temperature coil suction outlet [refrigerant vapor superheat at coil suction outlet is 6°F (3.3°C) or higher]
Entering liquid	35°F to 65°F (2°C to 18°C) temperature
Water flow rate	1.2 to 6 gpm per ton (0.02 to 0.1 L/s per kW) [equivalent to a water temperature rise of from 4°F to 20°F (2°C to 11°C)]
Water velocity	1 to 8 fps (0.3 to 2.4 m/s)

2.6 Determining Refrigeration Load

The following method of calculating refrigeration load shows a division of the true sensible and latent heat loss of the air which is accurate within the limitations of the data. This division does not correspond to load determination obtained from approximate factors or constants.

The total refrigeration load q_t of a cooling and dehumidifying coil (or air washer) per unit mass dry air is indicated in Figure 4 and consists of the following components:

1. The sensible heat q_s removed from the dry air and moisture in cooling from entering temperature t_1 to leaving temperature t_2 .
2. The latent heat q_l removed to condense the moisture from W_1 to W_2 .
3. The heat leaving the system as liquid condensate.

Items 1, 2 and 3 may be related by

$$q_t = q_s + q_l - q_w \quad (1)$$

Table 1 Example of Cooling Coil Catalog Data

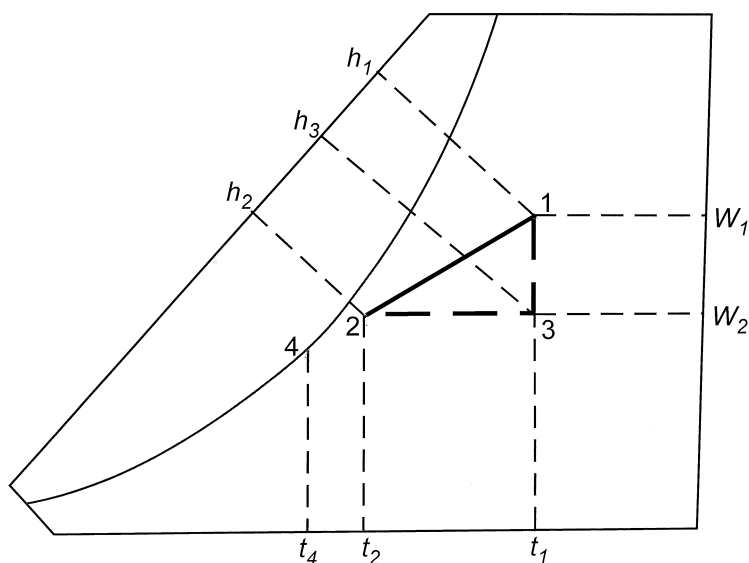
44EWT

EDB/EWB=80/67

WTR=8

WTR=10

FPS	ROW	FFP	400 FPM												500 FPM												600 FPM												700 FPM												FFP	ROW	FPS
			MBH	LDB	LWB	MBH	LDB	LWB	MBH	LDB	LWB	MBH	LDB	LWB	MBH	LDB	LWB	MBH	LDB	LWB	MBH	LDB	LWB	MBH	LDB	LWB	MBH	LDB	LWB																								
2	2	80	7.1	66.1	61.6	7.7	67.3	62.4	8.3	68.1	62.9	8.8	68.7	63.3	6.8	66.3	61.8	7.5	67.4	62.5	8.0	67.9	63.0	8.5	69.0	63.4	80	2	80																								
		100	8.1	64.3	60.9	8.8	65.4	61.7	9.4	66.4	62.3	10.0	67.1	62.7	7.8	64.5	61.1	8.5	65.6	61.9	9.1	66.2	62.4	9.7	67.5	62.9	100		100																								
		120	8.8	62.9	60.3	9.6	64.1	61.1	10.3	65.1	61.8	10.9	65.9	62.3	8.5	63.2	60.5	9.3	64.4	61.4	10.0	64.9	62.0	10.6	66.3	62.5	120		120																								
		140	9.3	62.0	59.8	10.2	63.2	60.8	11.0	64.2	61.4	11.6	65.0	62.0	9.0	62.2	60.1	9.9	63.4	61.0	10.6	64.0	61.6	11.3	65.4	62.2	140		140																								
		160	9.8	61.2	59.4	10.7	62.5	60.4	11.5	63.5	61.2	12.2	64.3	61.7	9.5	61.5	59.7	10.4	62.7	60.7	11.1	63.2	61.4	11.8	64.7	61.9	160		160																								
		80	12.1	59.0	57.4	13.5	60.4	58.6	14.6	61.6	59.5	15.6	62.6	60.2	11.7	59.3	57.8	13.0	60.8	58.9	14.1	61.9	59.8	15.0	62.8	60.4	80	4	80																								
		100	13.5	57.2	56.3	15.1	58.6	57.5	16.4	59.8	58.5	17.5	60.8	59.3	13.0	57.6	56.7	14.5	59.0	57.9	15.8	60.1	58.8	16.9	61.1	59.6	100		100																								
		120	14.4	56.0	55.4	16.2	57.4	56.8	17.6	58.6	57.8	18.8	59.6	58.6	13.9	56.4	55.9	15.6	57.8	57.1	17.0	59.0	58.1	18.2	59.9	58.9	120		120																								
		140	15.1	55.1	54.8	17.0	56.6	56.2	18.5	57.8	57.3	19.8	58.8	58.1	14.6	55.6	55.2	16.4	57.0	56.8	17.9	58.2	57.6	19.2	59.1	58.4	140		140																								
		160	15.7	54.5	54.2	17.7	56.0	55.7	19.3	57.2	56.8	20.6	58.3	57.8	15.2	55.0	54.7	17.1	56.4	56.1	18.6	57.6	57.2	19.9	58.6	58.1	160		160																								
	4	80	15.8	54.7	54.2	17.8	56.3	55.6	19.5	57.5	56.7	21.1	58.6	57.5	16.6	53.6	53.4	18.9	55.1	54.8	20.8	56.3	55.9	22.5	57.3	56.8	80	2	80																								
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		140	18.9	51.4	51.3	21.6	52.9	52.8	23.9	54.2	54.0	25.9	55.2	55.1	18.9	51.4	51.3	21.7	52.9	52.8	24.0	54.1	54.0	26.0	55.2	55.1	140		140																								
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		120	20.6	49.6	49.5	24.0	51.1	51.0	26.9	52.3	52.2	29.4	53.3	53.2	20.1	50.2	50.1	23.3	51.6	51.5	26.1	52.8	52.7	28.5	53.8	53.7	120		120																								
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		100	15.1	55.6	54.8	17.1	57.1	56.1	18.8	58.4	57.1	20.2	59.4	57.9	14.6	56.2	55.3	16.5	57.6	56.5	18.1	58.8	57.5	19.5	59.8	58.3	100		100																								
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		100	21.2	49.0	48.9	24.9	50.3	50.2	28.2	51.4	51.3	31.1	52.5	52.3	20.7	49.6	49.5	24.2	50.9	50.8	27.4	52.0	51.9	30.2	53.0	52.9	100		100																								
		120	22.1	48.1	48.0	26.2	49.3	49.2	29.8	50.4	50.3	33.1	51.4	51.3	21.6	48.6	48.5	25.6	49.8	49.7	29.0	50.9	50.8	32.1	51.9	51.8	120		120																								
		140	22.8	47.4	47.3	27.1	48.5	48.4	31.0	49.6	49.5	34.4	50.6	50.5	22.3	47.9	47.8	26.5	49.1	49.0	30.2	50.2	50.1	33.5	51.1	51.0	140		140																								
		160	23.2	46.9	46.8	27.8	48.0	47.9	31.8	49.0	48.9	35.4	50.0	49.9	22.8	47.4	47.3	27.2	48.5	48.4	31.0	49.6	49.5	34.5	50.6	50.5	160		160																								
4	2	80	8.5	64.9	60.5	9.3	66.2	61.4	10.1	67.1	61.9	10.7	67.9	62.4	8.1	65.2	60.8	9.0	66.4	61.6	9.7	67.4	62.2	10.3	68.1	62.6	80	2	80																								
		100	9.9	62.7	59.4	11.0	64.0	60.3	11.9	65.0	61.0	12.6	65.9	61.5	9.5	63.1	59.7	10.5	64.3	60.6	11.4	65.3	61.2	12.2	66.1	61.8	100		100																								
4	120	11.0	61.0	58.5	12.2	62.4	59.5	13.3	63.5	60.2	14.1	64.4	60.8	10.5	61.4	58.8	11.7	62.7	59.8	12.7	63.8	60.5	13.6	64.6	61.1	120	120																										
	140	11.9	59.8	57.7	13.2	61.2	58.8	14.3	62.3	59.6	15.3	63.2	60.3	11.4	60.2	58.1	12.7	61.5	59.1	13.8	62.6	59.9	14.7	63.5	60.6	140	140																										
6	160	12.6	58.7	57.0	14.1	60.2	58.2	15.3	61.4	59.1	16.3	62.3	59.8	12.1	59.2	57.5	13.5	60.6	58.6	14.7	61.7	59.4	15.7	62.6	60.1	160	160																										
	80	14.1	57.2	55.7	15.9	58.7	57.0	17.4	60.0	57.9	18.8	61.0	58.7	13.5	57.7	56.2	15.3	59.2	57.4	16.8	60.4	58.3	18.1	61.3	59.0	80	4	80																									
100	15.9	55.0	54.1	18.1	56.4	55.4	19.9	57.7	56.4	21.6	58.7	57.3	15.3	55.5	54.7	17.4	56.9	55.9	19.2	58.1	56.9	20.8	59.1	57.7	100	100																											
4	120	17.2	53.4	52.9	19.																																																



Psychrometric Performance of a
Cooling and Dehumidifying Coil

Fig. 4 Psychrometric Performance of Cooling and Dehumidifying Coil

If only the total heat value is desired, it may be computed by

$$q_t = (h_1 - h_2) - (W_1 - W_2)h_{f4} \quad (2)$$

where

h_1 and h_2 = enthalpy of moist air at points 1 and 2, respectively

W_1 and W_2 = humidity ratio at points 1 and 2, respectively

h_{f4} = enthalpy of saturated liquid at final temperature

$W_1 - W_2$ = mass of water vapor condensed per unit mass of air

If a breakdown into latent and sensible heat components is desired, the following relations may be used. The latent heat may be found from

$$q_l = (h_1 - h_3) - (W_1 - W_2)h_{f2} \quad (3)$$

where

h_{g1} = enthalpy of saturated water vapor at temperature t_1

h_{g3} = enthalpy defined by temperature T_1 , and humidity ratio W_2

The sensible heat may be shown to be

$$q_s = (h_3 - h_2) + (W_1 + W_2)(h_{f2} - h_{f4}) \quad (4)$$

The final condensate temperature t_4 leaving the system is subject to substantial variations, depending on the method of coil installation, as affected by such factors as coil face orientation, airflow direction and air duct insulation. In practice, t_4 is frequently the same as the leaving wet-bulb temperature. Within the normal air-conditioning range, precise values of t_4 are not necessary since energy in the condensate removed from the air usually represents about 0.5 to 1.5% of the total refrigeration load. Values needed to calculate moist air properties can be found on the ASHRAE Psychrometric Chart (Chart 1) and Tables 2 and 3 of Chapter 1 of the 2021 *ASHRAE Handbook—Fundamentals*.

Example 1 Air enters a coil at 90°F DB, 75°F WB; it leaves at 61°F DB, 58°F WB; leaving liquid water is at 54°F.

Solution

From the ASHRAE Psychrometric Chart, find the following:

$$\begin{aligned}h_1 &= 38.37 \text{ Btu/lb}_m \text{ dry air} \\h_2 &= 25.06 \text{ Btu/lb}_m \text{ dry air} \\h_3 &= 32.14 \text{ Btu/lb}_m \text{ dry air} \\W_1 &= 106.63 \text{ grains water vapor/lb}_m \text{ dry air} \\W_2 &= 67.04 \text{ grains water vapor/lb}_m \text{ dry air}\end{aligned}$$

From Table 3 of Chapter 1 of the 2021 *ASHRAE Handbook—Fundamentals*, find the following:

$$h_{f4} = 22.07 \text{ Btu/lb}_m \text{ liquid water}$$

From Equation (2), the total heat is:

$$\begin{aligned}q_t &= (h_1 - h_2) - (W_1 - W_2)h_{f4} \\&= (38.37 - 25.06) - \frac{(106.63 - 67.04)}{7000}(22.07) \\&= 13.19 \text{ Btu/lb}_m \text{ dry air}\end{aligned}$$

From Equation (3), the latent heat is

$$\begin{aligned}q_1 &= (h_1 - h_3) - (W_1 - W_2)h_{f2} \\&= (38.37 - 32.14) - \frac{(106.63 - 67.04)}{7000}29.08 \\&= 6.07 \text{ Btu/lb}_m \text{ dry air}\end{aligned}$$

From Equation (4), the sensible heat is

$$\begin{aligned}q_s &= (h_3 - h_2) + (W_1 - W_2)(h_{f2} - h_{f4}) \\&= (32.14 - 25.06) + \frac{(106.63 - 67.04)}{7000}(29.08 - 22.07) \\&= 7.12 \text{ Btu/lb}_m \text{ dry air}\end{aligned}$$

The heat content of the leaving condensate is

$$\begin{aligned}q_w &= (W_1 - W_2)h_{f4} \\&= \frac{(106.63 - 67.04)}{7000}22.07 \\&= 0.12 \text{ (Btu/lb}_m \text{ dry air)}\end{aligned}$$

3 Heating Coils

Generally, extended-surface coils are used for heating air with steam or hot water.

3.1 Steam Coils

For proper performance of steam heating coils, condensate and air or other noncondensables must be rapidly eliminated and the steam must be uniformly distributed to the individual tubes. Noncondensable gases (such as air or carbon dioxide) that remain in the coil cause chemical corrosion and result in early coil failure.

Steam is distributed uniformly by methods such as:

- Individual orifices in the tubes
- Distributing plates in the steam headers
- Perforated, small-diameter, inner steam-distributing tubes extending to the larger tubes of the primary surface

Coils with perforated inner tubes are constructed with different arrangements such as:

- Supply and return on one end, with the incoming steam used to heat the leaving condensate
- Supply and return on opposite ends
- Supply and return on one end and a supply on the opposite end

Properly designed and selected steam distribution coils distribute steam throughout the full length of all primary tubes, even when the leaving air temperature is controlled by modulating the steam supply through a steam-metering valve. Thus, more uniform leaving air temperatures are produced over the entire length and face of the coil than from a single-tube coil.

Piping, controls, and installation must be designed to protect the coils from freezing due to incomplete draining of condensate. When the entering air temperature is 32°F (0°C) or below, the steam supply to the coil should not be modulated. A series of coils in the airstream, with each coil sized to be on or completely off in a specific sequence depending on the entering air temperature, has less likelihood for a freeze-up. Bypass dampers are also common. When less than full load conditions occur, air is bypassed around the steam coil with full steam being kept on the coil. In this system, the high-velocity jets of low-temperature air must not impinge on the coil when the face dampers are in a partially closed position.

3.2 Water Coils

In order to produce the desired heating capacity, hot water comfort heating systems usually require no more than one or two rows of tubes in the direction of airflow. Various circuits are used to produce the most efficient capacity without excessive water pressure drop through the coil. The relative directions of fluid flows influence the performance of heat transfer surfaces. In air-heating coils with only one row of tubes, the air flows at right angles relative to the heating medium. In coils with more than one row of tubes in the direction of the airflow, the heating medium in the tubes may be circuited in various ways.

Although crossflow is common in steam-heating coils, parallel flow and counterflow arrangements are common in water coils. Counterflow is the preferred arrangement to obtain the highest possible mean temperature difference.

A single-tube serpentine circuit on small-size booster heaters requires small water quantities up to a maximum of approximately 4 or 5 gpm (0.25 or 0.32 L/s). With this arrangement, a single tube handles the entire water quantity, provided the tube is circuited such that it makes a number of passes across the airstream.

Commonly selected water circuits are illustrated in Figure 5. When entering air temperatures are below freezing (an antifreeze brine is sometimes used), piping the coil for parallel flow rather than counterflow should be considered, placing the highest water temperature on the entering air side. Coils piped for counterflow have water enter the coil in the tube row on the exit air side of the coil. Coils piped for parallel flow have water enter the tube row on the entering air side of the coil.

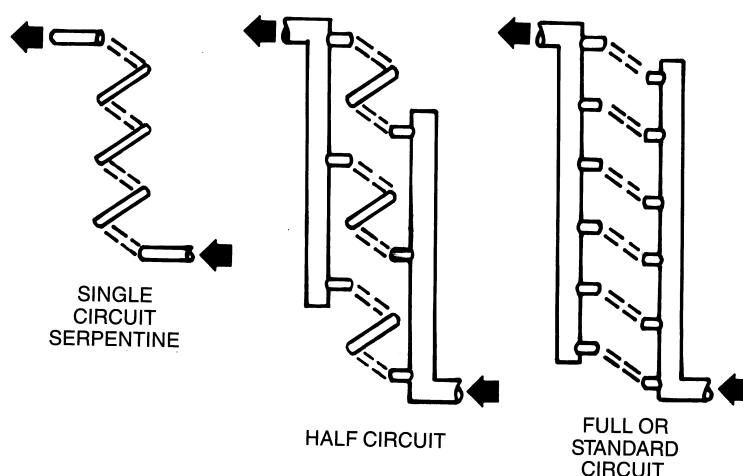


Fig. 5 Water Circuit Arrangements for Heating Coils

If air temperatures near or below freezing are anticipated, full design water flow rate should be ensured whenever the temperature approaches freezing. These coils are usually provided with a coil pump that provides constant flow through the pump with varying water temperature for reduced load control.

3.3 Electric Heating Coils

An electric heating coil consists of a length of resistance wire (commonly nickel/chromium) to which a voltage is applied. The resistance wire may be bare or sheathed. The sheathed coil is a resistance wire encased by an electrically insulating layer such as magnesium oxide, which is encased in a finned steel tube. The sheathed coils are more expensive, have a higher air-side pressure drop, and require more space. The outer surface temperature of sheathed coils is lower, the coils are mechanically stronger, and electrical contact with body or housing is prevented.

3.4 Coil Ratings

Steam and hot water coils are usually rated within these limits, which may be exceeded for special applications:

Air face velocity	Between 200 and 1500 fpm (1 to 8 m/s), based on air at standard density of 0.075 lb/ft ³ (1.2 kg/m ³)
Entering air	−20°F to 100°F (−30°C to 38°C) for steam temperature coils; 0°F to 100°F (−20°C to 38°C) for hot water coils
Steam pressures	From 2 to 250 psia (14 to 1700 kPa) at the coil steam supply connection (pressure drop through the steam control valve must be considered)
Hot water	Between 120°F and 250°F (50°C and 120°C) temperature
Water velocities	From 0.5 to 8 fps (0.2 to 2.5 m/s)

Individual installations vary widely, but the following values can be used as a guide. The most common air face velocities used are between 500 and 1000 fpm (2.5 and 5 m/s). Delivered air temperatures vary from about 72°F (22°C) for ventilation only to about 150°F (66°C) for complete heating. Steam pressures vary from 2 to 15 psig [15 to 100 kPa (gage)], with 5 psig [35 kPa (gage)] being the most common. A minimum steam pressure of 5 psig [35 kPa (gage)] is recommended for systems with entering air temperatures below freezing. Water temperatures for comfort heating are commonly between 120 and 200°F (50 and 93°C), with water velocities between 4 and 6 fps (1.2 and 1.8 m/s). Water quantity is usually based on about 20 to 40°F (10 to 20°C) temperature drop through the coil. Air resistance is usually limited to 0.4 to 0.6 in. (100 to 150 Pa) of water for commercial buildings and to about 1 in. (250 Pa) for industrial buildings. High-temperature water systems have water temperatures commonly between 300 and 400°F (150 and 200°C), with up to 100°F (55°C) drops through the coil.

4 Evaporative Air-Cooling Equipment

In its broadest sense, the evaporative cooling principle applies to all equipment that exchanges sensible heat for latent heat. Cooling towers, evaporative condensers, vacuum cooling apparatus, air washers, spray coil dehumidifiers, and packaged evaporative coolers cool air by evaporation. This equipment falls into two general categories: (1) apparatus for air cooling and (2) apparatus for heat rejection.

Evaporative air cooling evaporates water into an airstream. Illustrated in Figure 6 are thermodynamic changes that take place between air and water that are in direct contact in the moving airstream. The continuously recirculated water has reached an equilibrium temperature that equals the entering air wet-bulb temperature. The heat and mass transfer process between the air and water lowers the air dry-bulb temperature and increases the humidity ratio at constant wet-bulb temperature.

The extent to which the leaving air temperature approaches the thermodynamic wet-bulb temperature of the entering air, or the extent to which complete saturation is approached, is conveniently expressed as cooling or saturation efficiency and is defined as

$$e_c = (t_1 - t_2)/(t_1 - t') \quad (5)$$

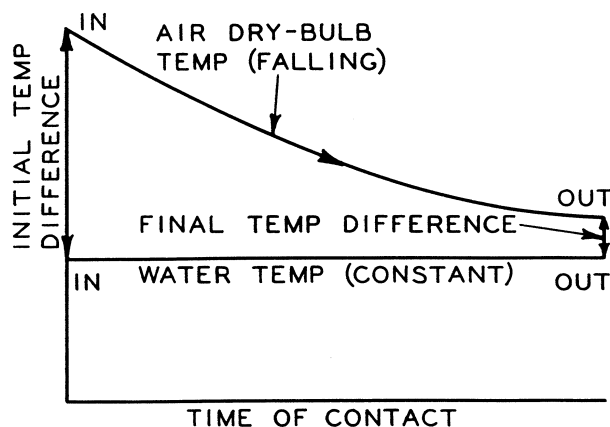


Fig. 6 Interaction of Air and Water in Evaporative Air Coolers

where

- e_c = cooling or saturation efficiency
- t_1 = dry-bulb temperature of the entering air
- t_2 = dry-bulb temperature of the leaving air
- t' = thermodynamic wet-bulb temperature of entering air

If warm or cold unrecirculated water is used, the air can be heated and humidified or cooled and dehumidified.

Evaporative air-cooling equipment can be placed in two general classes, direct and indirect. In the direct system, air is cooled by direct contact with the water, either from the wetted surface of an extended-surface material (as in packaged air coolers) or with a series of sprays (as in an air washer). In the indirect system, air is cooled in a heat exchanger by a secondary stream of air and water that has been evaporatively cooled, such as by a cooling tower and cooling coil.

By applying recirculating, regenerating principles, temperatures below the initial wet bulb may be produced. However, the cost of these more complex devices has restricted their use.

5 Air Washers

A spray-type air washer consists of a chamber or casing containing a spray nozzle system, a tank for collecting the spray water as it falls, and an eliminator section at the discharge to remove the entrained drops of water from the air. A pump recirculates water at a rate in excess of the evaporation rate. Heat and mass transfer between the air and the water cools the water.

Construction features of conventional spray-type air washers are shown in Figure 7. Requirements of air washer operation are:

- Uniform distribution of air across the spray chamber
- Air velocities from 300 to 700 fpm (1.5 to 3.5 m/s) in the washer chamber
- Adequate spray water broken up into fine droplets at pressures from 20 to 40 psig [140 to 280 kPa (gage)]
- Good spray distribution across the airstream
- Sufficient length of travel through the spray and wetted surfaces
- Elimination of free moisture from the outlet air

Spray water requirements for spray-type air washers used for washing or evaporative cooling vary from 4 gpm per 1000 cfm with a single bank to 10 gpm per 1000 cfm for double banks (0.5 to 1.3 L/s per m³/s). Pumping heads usually range from 55 to 150 ft (160 to 450 kPa) of water, depending on such factors as spray pressure, height of apparatus, and pressure losses in pipe and strainers.

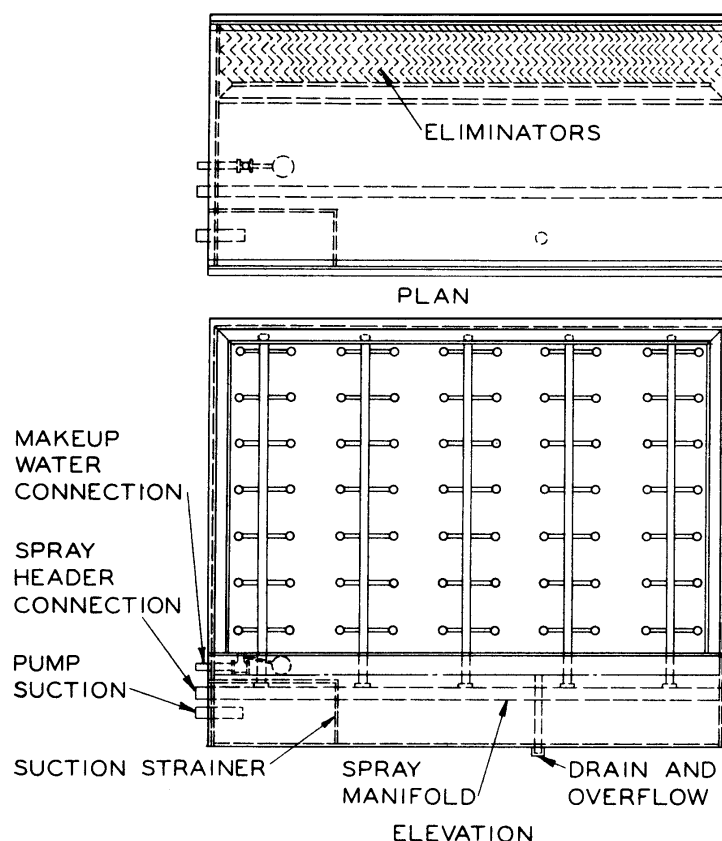


Fig. 7 Single-Bank Air Washer

6 Dehumidification

Dehumidification is the reduction of the water content of air, gases, or other fluids. Dehumidification is normally limited to equipment operating at essentially atmospheric pressures, built to standards similar to other types of air-handling equipment.

Drying of gases has become an increasingly important operation. Some commercial applications include the following:

- Lowering the relative humidity to facilitate manufacturing and handling of hygroscopic materials
- Air conditioning for comfort (in combination with cooling, under certain design conditions, such as high moisture load in comparison to sensible cooling load)
- Providing protective atmospheres for the heat treatment of metals
- Maintaining controlled humidity conditions in warehouses and caves for storage
- Preserving ships and other surplus equipment, which would otherwise deteriorate
- Condensation and corrosion control
- Drying air for wind tunnels
- Drying natural gas
- Drying instrument air and plant air
- Drying of process and industrial gases
- Dehydration of liquids

6.1 Methods of Dehumidification

Dehumidification can be accomplished by compression, refrigeration, liquid sorption, solid sorption, or combinations of these systems.

Three methods by which sorbent materials or sorbent equipment dehumidify air are illustrated in the skeleton psychrometric chart shown in Figure 8. Air at point A can be dehumidified and cooled to point B by a

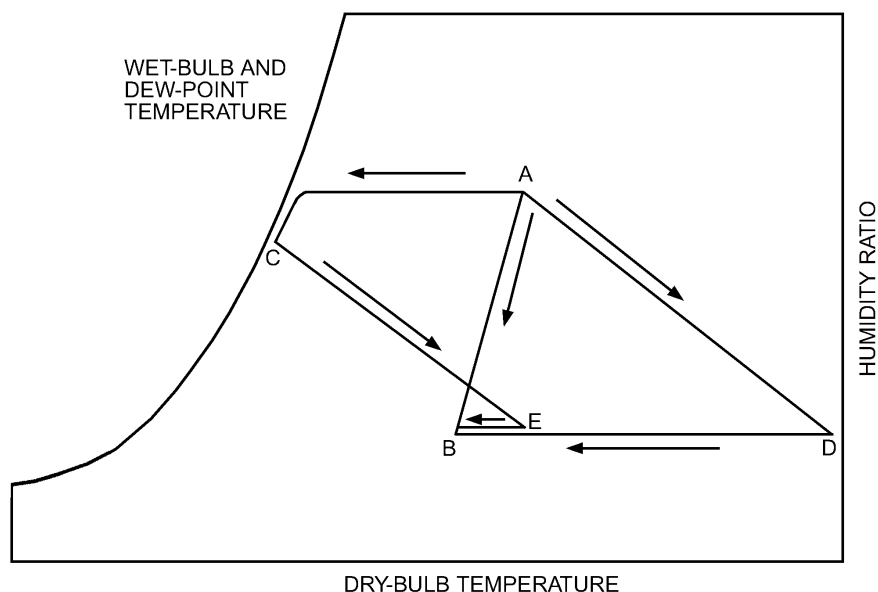


Fig. 8 Methods of Dehumidification
(Figure 1, Chapter 24, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

liquid sorption system with intercooling directly. Alternatively, it may be dehumidified in a solid sorption unit by precooling and dehumidifying from point A to point C, desiccating from point C to point E, and finally cooling to point B. Air could also be dehumidified with solid sorption equipment by desiccating from point A to point D and then by refrigeration from point D to point B.

Compression Dehumidification. Compressing a gas to be dehumidified reduces its absolute moisture content, but it generally produces a saturated condition at the elevated pressure. This method is uneconomical but is of value for pressure systems since part of the moisture is removed by compression of the gas; the remaining moisture may be removed by cooling alone, with sorption, or both, depending on the final dew point required. Expansion of high-pressure gas also lowers the dew point.

Refrigeration Dehumidification. Refrigerating gas below its dew point is the most common method of dehumidification. This method is advantageous when the gas (1) is comparatively warm, (2) has a high moisture content, and (3) requires an outlet dew point above 40°F (5°C). Frequently, refrigeration is used in combination with sorption dehumidifiers to obtain an extremely low dew point that is difficult to achieve with refrigeration alone.

Direct Sorption Dehumidification. Sorbent materials used in dehumidification equipment may either be liquids or solids. The performance of the different sorption dehumidification machines is, to some extent, a function of the sorbent used. Sorbents either retain water on their surface (adsorption) or chemically combine with water (absorption). In regenerative equipment, the water must be removed from the sorbent to regenerate it for further use.

Nonregenerative equipment uses hygroscopic salts such as calcium chloride, urea, or sodium chloride. In regenerative systems, the sorbent is usually a form of silica, alumina gel, activated alumina, molecular sieves, lithium chloride salt, lithium chloride solution, or glycol solution.

In liquid sorption dehumidification systems, the gas passes through sprays of a liquid sorbent such as lithium chloride or a glycol solution. The sorbent in active state has a vapor pressure below that of the gas to be dehumidified and absorbs moisture from the gas stream. The sorbent solution during the process of absorption becomes diluted with moisture, which during regeneration is given up to an outdoor airstream in which the solution is heated. A partial bleed-off of the solution is used for continuous reconcentration of the sorbent in a closed circuit between the spraying or contactor unit and the regenerator unit.

In solid sorption, the gas stream passes through or over granular beds of fixed desiccant structures. A number of commercially available desiccants may be used, depending on such factors as the character of the gas to be dried, inlet temperature, moisture levels, and required final dew point. Outdoor air is passed through

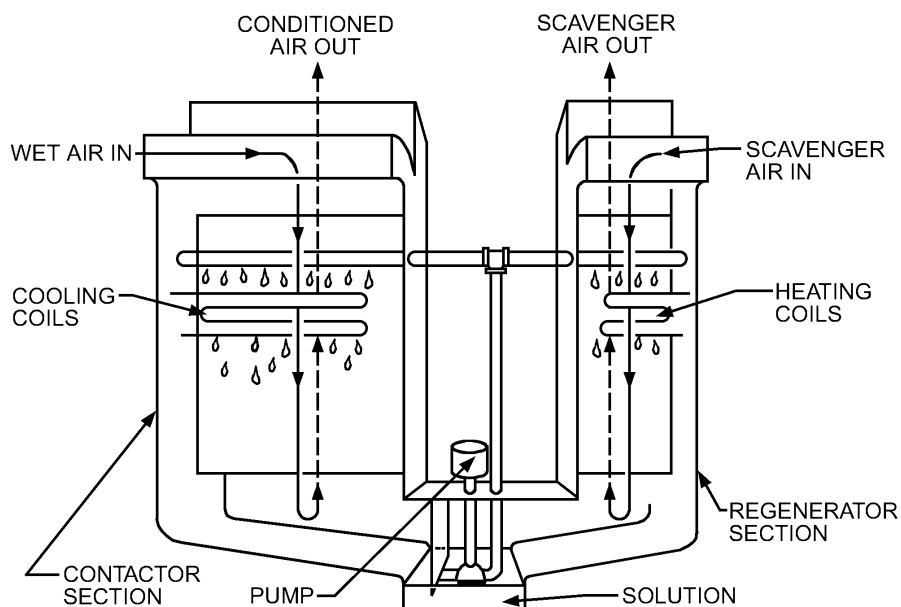


Fig. 9 Flow Diagram for Liquid-Absorbent Dehumidifier
(Figure 2, Chapter 24, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

beds or layers of the sorbent, which in its active state has a vapor pressure below that of the gas to be dehumidified and absorbs moisture from the gas stream. After becoming saturated with moisture, the desiccant is periodically reactivated to give up previously absorbed moisture to an outdoor air or gas stream.

6.2 Liquid Absorption Equipment

The flow diagram for a typical liquid absorption system with extended-surface contactor coils is shown in Figure 9. For dehumidifying operation, the strong absorbent solution is pumped from the sump of the unit and sprayed over the contactor coils.

Air to be conditioned passes over the contactor coils and comes in contact with the hygroscopic solution. Airflow can be either parallel with or counter to the sprayed solution flow, depending on the space and application requirements.

The degree of dehumidification depends on the concentration, temperature, and characteristics of the hygroscopic solution. Moisture is absorbed from the air by the solution due to the vapor pressure difference between the air and the liquid absorbent. The moisture content of the outlet air can be precisely maintained by varying the coolant flow in the coil to control the absorbent solution contact temperature. The absorbent solution is maintained at the proper concentration by continuous regeneration.

The heat generated in absorbing moisture from the air consists of the latent heat of condensation of water vapor and the heat of solution, or the heat of mixing, of the water and the absorbent. The heat of mixing varies with the liquid absorbent used and with the concentration and temperature of the absorbent. The total heat removal required by the conditioner coil consists of the heat of absorption, sensible heat removed from the air, and the residual heat load added by the regeneration process.

6.3 Solid Sorption

Dehumidification by use of a solid desiccant, such as silica gel, molecular sieves, activated alumina, or hygroscopic salts, may be performed under either static or dynamic conditions. In the static method, the air or gas is not forced through the desiccant. Instead, the air immediately surrounding the desiccant dries, and through convection and vapor diffusion, water vapor from surrounding areas reaches the desiccant where it is absorbed. In dynamic dehumidification, the air or gas being dried flows through the desiccant bed or structure. In the air-conditioning industry, which is primarily concerned with operation at atmospheric pressure, an air-moving device such as a fan forces the gas through the desiccant bed and a heater or other means periodically reactivates the desiccant.

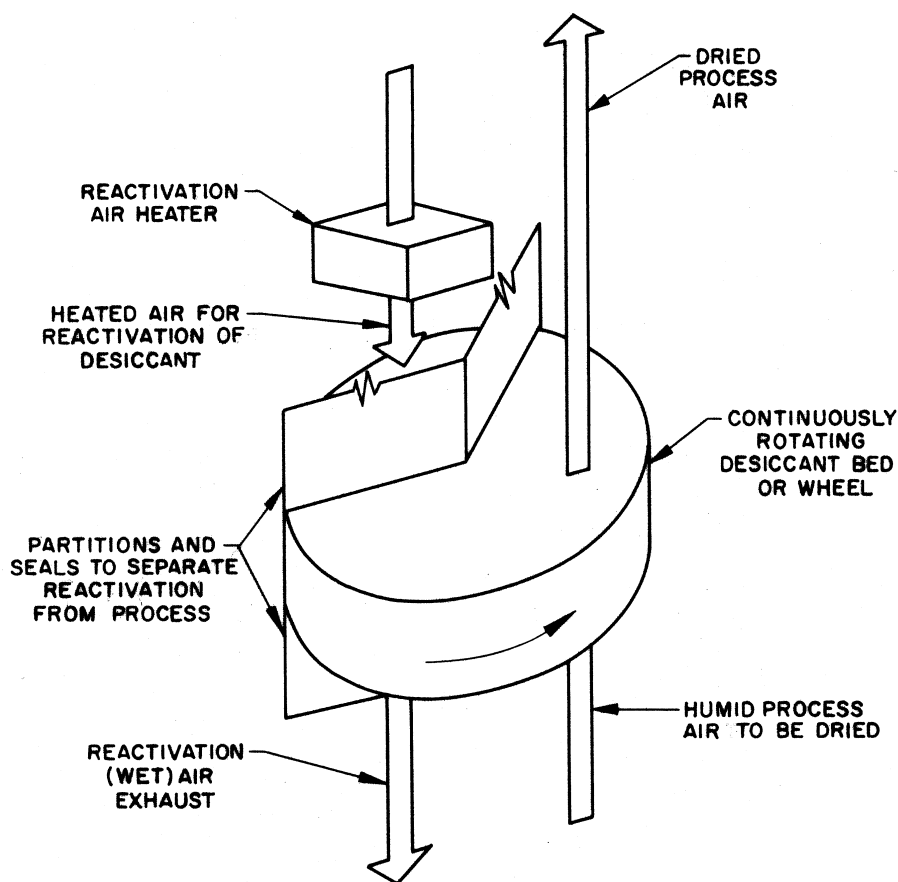


Fig. 10 Rotary Dehumidification Unit

6.4 Solid Absorption Equipment

The arrangement of major components of a typical solid absorption system is shown in Figure 10. To achieve dehumidification, moist air is passed through a desiccant structure, where the water vapor in the air is absorbed by the desiccant. The systems employ a desiccant structure that may be a disc, drum, or wheel, filled or saturated with an absorbent, such as lithium chloride. The desiccant structure rotates slowly through a heated stream of air where the desiccant is desorbed or reactivated. By continuous rotation, freshly reactivated portions of the desiccant are always available for drying.

The amount of drying depends on the temperature and absolute humidity of the air and the useful concentration of desiccant. The useful concentration of the desiccant is affected by:

- Quantity of desiccant in relation to the mass flow of air and water vapor
- Energy (amount of heat and temperature) for reactivation
- Frequency of reactivation (speed of rotation)

Performance typical of a solid absorption dehumidifier using a fixed desiccant structure is shown in Figure 11.

7 Humidification

When selecting humidification equipment, both the environmental conditions of the occupancy or process and the characteristics of the building enclosure must be considered. These may not always be compatible and a compromise solution may be necessary.

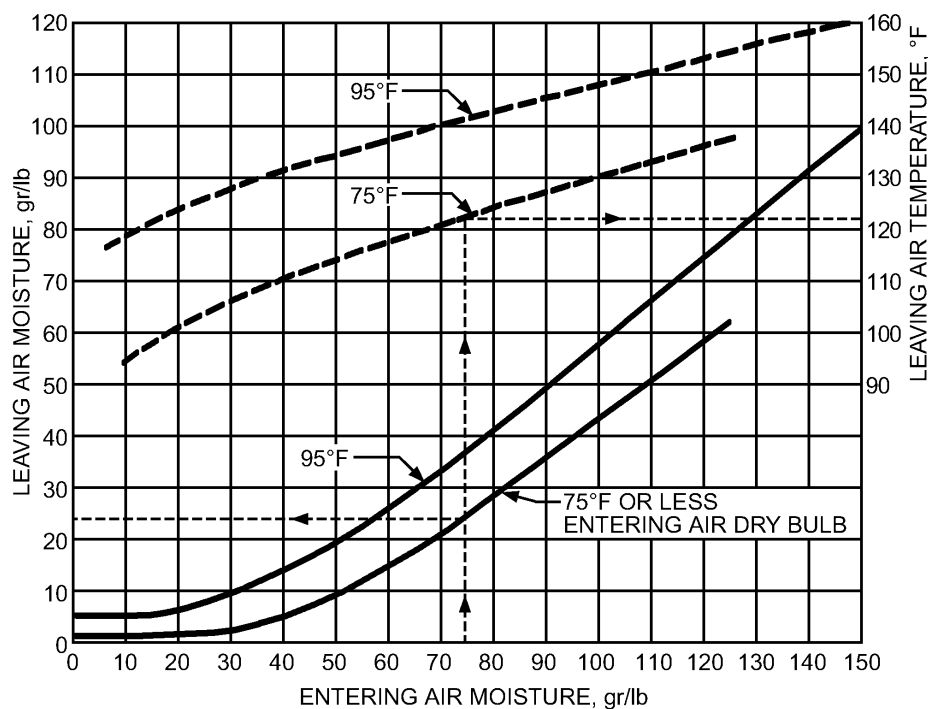


Fig. 11 Performance Data for Rotary Solid Absorption Dehumidifier
(Figure 13, Chapter 24, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

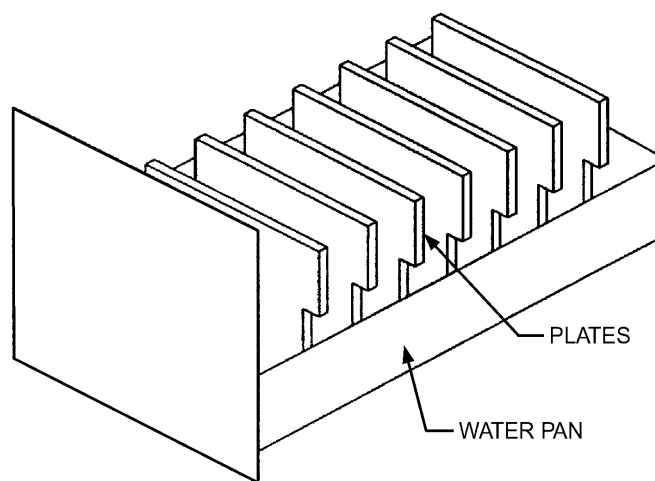


Fig. 12 Pan-Type Humidifier

Air washers and evaporative coolers may be used as humidifiers but are usually selected to provide some additional function, such as air cooling or air cleaning, as discussed in the section on air washers.

Residential Humidifiers for Central Air Systems. This type of unit depends on airflow in the heating system for evaporation and distribution. General principles of operation and description of equipment are as follows:

Pan Type. The humidification rate varies with temperature, humidity, and velocity of the air in the system (Figure 12).

Wetted Elements. These units use an open textured, wetted media, through or over which air is circulated. The evaporating surface may take the form of a fixed pad, wetted by sprays or by water flowing through

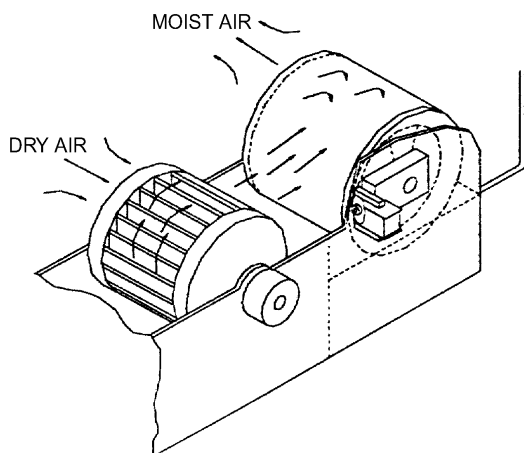


Fig. 13 Wetted-Drum Humidifier

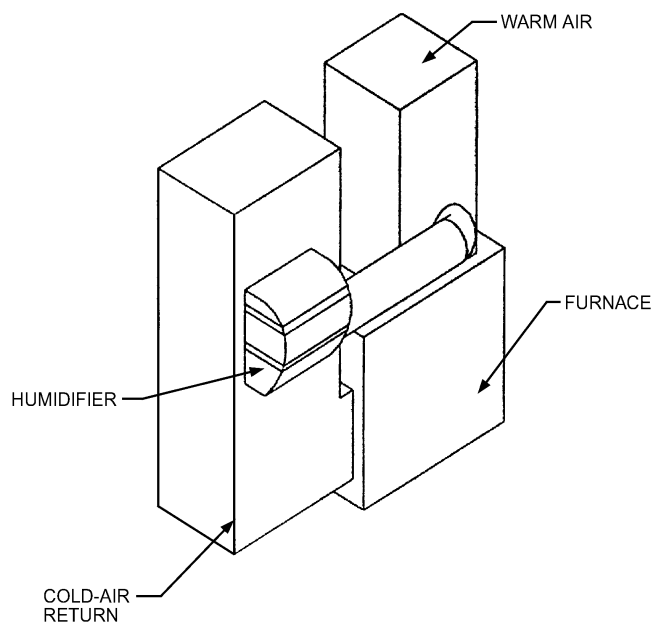


Fig. 14 Bypass Wetted-Element Humidifier

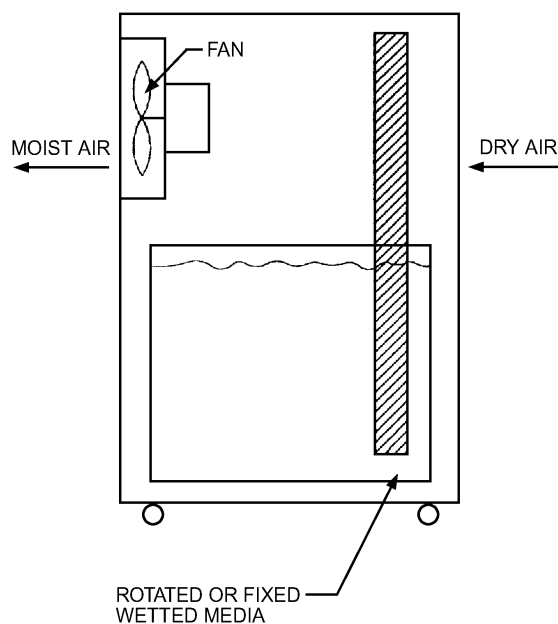


Fig. 15 Portable Humidifier

by gravity from a header at the top, or the pad may be a paddle wheel, drum, or belt rotating through a water reservoir. Figures 13 and 14 are examples of this type of humidifier.

Residential Humidifiers for Nonducted Applications. Many portable or room humidifiers are available for use in residences and apartments heated by nonducted systems, such as hydronic or electric, or where the occupant is prevented from making a permanent installation. An example of this type of humidifier is shown in Figure 15.

Industrial/Commercial Humidifiers for Central Air Systems. Humidification equipment commonly used in central air-handling systems incorporates a heated water pan, direct steam injection, or atomizers. Specific types are shown in Figure 16 and discussed briefly in the following paragraphs.

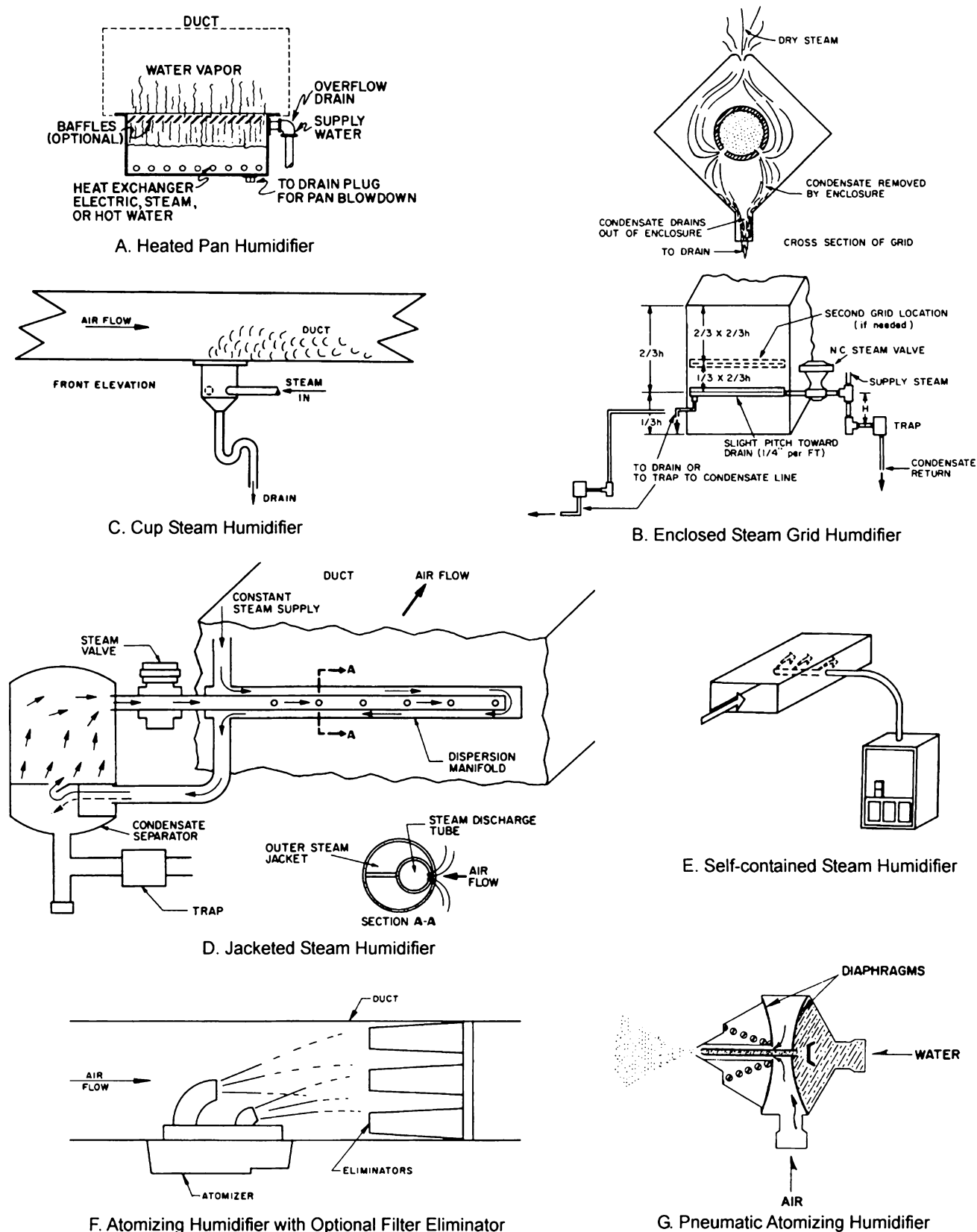


Fig. 16 Commercial Humidifiers

Heated Pan. These units offer a broad range of capacity and may be heated by an electric element, steam, or hot water coil (see Figure 16A). Some units are designed to be attached directly to the underside of a system duct. Others are provided with a fan or a steam hose, which allows them to be installed remote from the ductwork.

Steam. Direct steam injection humidifiers cover a wide range of designs and capacities. Since water vapor is steam at low pressure and temperature, the whole process can be simplified by introducing steam directly into the air to be humidified. This method is essentially an isothermal process because the temperature of the air remains constant as the moisture is added in vapor form. The steam control valve may be modulating or two-position in response to a humidity controller. The steam may be from an external source, as in the enclosed grid, cup, or jacketed dry steam humidifiers, or produced within the humidifier, as in the self-contained type.

- *Enclosed steam grid humidifiers* (Figure 16B) should be used on low steam pressures to prevent splashing of condensate in the duct.
- A *cup or pot-type humidifier* (Figure 16C) is usually attached under a system duct. Steam is attached tangentially to the inner periphery of the cup by one or more steam inlets, depending on the capability of the unit. The steam supply line should have a suitable steam trap.
- A *jacketed steam humidifier* uses an integral steam valve with a steam-jacketed duct-traversing dispersing tube and condensate separator to prevent condensate from being introduced into the airstream (Figure 16D). An inverted bucket-type steam trap is required to drain the separating chamber.

The aforementioned humidifiers inject steam directly from the boiler into the space or duct system. Some boiler treatment chemicals can be discharged, which can affect indoor air quality.

- A *self-contained steam humidifier* converts tap water to steam by electrical energy using either the electrode boiler principle or resistance heating. This steam is injected into the duct system through a dispersion manifold (Figure 16E), or the humidifier may be freestanding for nonducted applications.

Atomizing Humidifiers with optional filter eliminator (Figure 16F). Centrifugal atomizers use a high-speed disc that slings water through a fine comb to create a fine mist that is introduced directly into the air where it is evaporated. The ability of the air to absorb the moisture depends on temperature, air velocity, and moisture content. Where mineral fallout from hard water is a problem, optional filter eliminators may be added to remove mineral dust from humidified air, or water demineralizers may be installed. Additional atomizing methods use nozzles; one uses water pressure and the other uses both air and water, as shown in Figure 16G. Mixing air and water streams at combined pressures atomizes water into a fine mist, which is evaporated in the room or air duct.

Wetted-Element Humidifier. Wetted-element humidifiers have a wetted media, sometimes in modular configurations, through or over which air is circulated to evaporate water. This unit depends on airflow for evaporation; the rate varies with temperature, humidity, and velocity of the air.

8 Sprayed Coil Humidifiers/Dehumidifiers

A special adaptation of an air washer can be used very effectively to simultaneously control both the temperature and humidity of an outdoor airstream as in a dedicated outdoor air system (DOAS).

The cooling coil system provided with a deep cooling coil (usually 8 to 12 rows with flat [nonenhanced] fins spaced not less than 1/8 in. apart and with the entire coil assembly and housing made of a noncorrosive material such as stainless steel.) (Figure 17a).

The coil is located in a stainless steel section of the air-handling unit over a drain pan with an air spray and recirculating pump. The air spray sprays water from the pump into the airstream against the cooling coil. The water tends to approach the average temperature of the coil and the leaving airstream dew-point temperature. The coil/spray assembly essentially has a coil efficiency of 100% (bypass factor of 0), and the coil-leaving air is saturated at the temperature of the water.

Since the system also functions as an air washer, it very effectively removes particulate (and some chemical) contamination from the air.

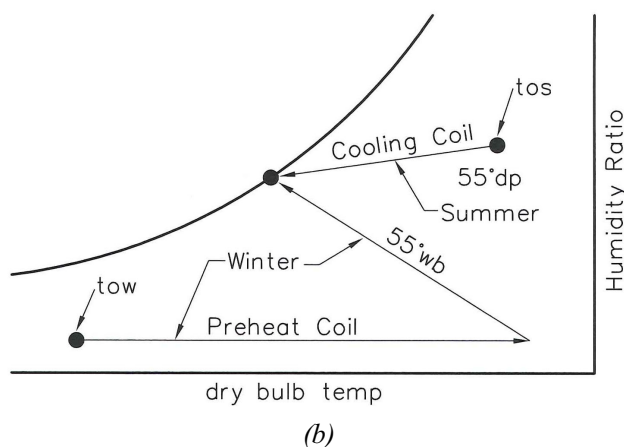
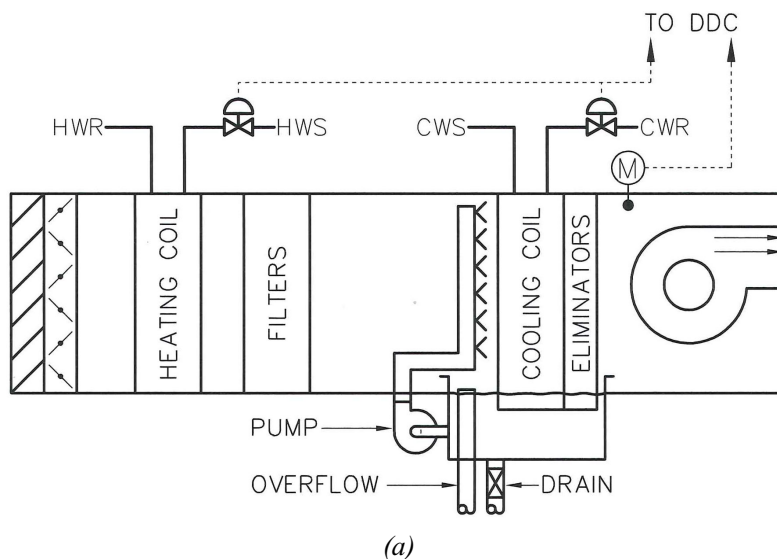


Fig. 17 (a) Sprayed Coil DOAS and (b) Psychrometric Process for Sprayed Coil DOAS

Another benefit is that it serves as both a humidifier and a dehumidifier as necessary, depending upon the dew-point temperature or humidity ratio of the leaving air stream compared to the entering air stream. Figure 17b is a skeleton psychrometric chart illustrating the performance as a humidifier and as a dehumidifier. It is controlled by a single discharge temperature sensor set at the desired dew point temperature and operating the heating coil and chilled water coil in sequence. For freeze protection, the heating coil is usually equipped with a constant flow pump or an integrated face and bypass damper.

The sprayed coil DOAS is very effective in maintaining a fixed dew point temperature for critical applications such as art museums, archived storage, rare book libraries, critical humidity manufacturing processes, surgical suites, etc.

Since the sprayed coil and water system functions as an air washer, it requires scheduled blow down and cleaning to remove the particulate collected from the pan and the coil surfaces. Also, to prevent bacterial growth, the safety controls should shut the system down if the water temperature exceeds 58°F.

9 Air Cleaners

In buildings, dust content is usually less than 0.2 mg/m^3 ($0.01 \times 10^{-6} \text{ lb}_m/\text{ft}^3$) of air. This concentration is in contrast to exhaust gases from processes and flue gases where dust concentration typically ranges from 200 to 40 000 mg/m^3 (10×10^{-6} to $2000 \times 10^{-6} \text{ lb}_m/\text{ft}^3$).

With certain exceptions, the air cleaners described in this section are not applicable to the cleaning of exhaust gases. Atmospheric dust is a complex mixture of smokes, mists, fumes, dry granular particles, and fibers (such particles, when suspended in a gas, are called aerosols). A sample of atmospheric dust gathered at any point generally contains materials common to that locality, together with other components that originated at a distance but have been transported by air currents or diffusion.

These components vary with the geography of the locality in question, the season of the year, the direction and strength of the wind, and the proximity of dust sources. A sample of atmospheric dust usually contains soot and smoke, silica, clay, decayed animal and vegetable matter, organic materials in the form of lint and plant fibers, and metallic fragments. It may also contain living organisms, such as mold spores, bacteria, and plant pollens, which may cause diseases or allergic responses. Chapter 11 of the 2021 *ASHRAE Handbook—Fundamentals* contains further information on atmospheric contaminants.

The particles in the atmosphere can range in size from less than 0.01 mm to dimensions of lint, leaves, and insects. Almost all conceivable shapes and sizes are represented. This wide variety makes it impossible to design one cleaner that is best for all applications. For example, in industrial ventilation, only the coarser dust particles may need to be removed from the airstream for cleanliness of the structure and protection of mechanical equipment. In other applications, surface discoloration must be prevented. Unfortunately, the smaller components of atmospheric dust are the worst offenders in smudging and discoloring building interiors. Electronic air cleaners or high-efficiency dry filters are required for small particle removal. In clean room applications or when radioactive or other dangerous particles are present, extremely high-efficiency mechanical filters should be used.

The most important characteristics of aerosols affecting the performance of an air filter include particle size and shape, density, and concentration. The most important of these is particle size. Data on the sizes and characteristics of airborne particulate matter and the wide range of particle size encountered are given in Figure 18. Cleaning efficiency is also affected by the velocity of the airstream. The degree of air cleanliness required is a major factor influencing filter design and selection. Removal of particles becomes progressively more difficult as particle size decreases.

Cost considerations (both in initial investment and maintenance), space requirements, and airflow resistance, in addition to wide-ranging criteria as to degree of air cleanliness, have resulted in a wide variety of commercial air cleaners.

To evaluate filters and air cleaners properly for a particular application, the following factors should be considered:

- Degree of air cleanliness required
- Disposal of dust after it is removed from the air
- Amount and type of dust in the air filtered
- Operating resistance to airflow (pressure drop)
- Space available for filtration equipment
- Cost of maintaining or replacing filters
- Initial cost of the system

Figure 19 shows typical locations at filters and lists some applications of filters classified according to their efficiencies and type.

9.1 Types of Air Cleaners

Air cleaners fall into the following three categories:

1. *Fibrous media unit filters* in which accumulating dust load causes pressure drop to increase up to some maximum permissible value. During this period, efficiency normally increases; however, at high dust loads, dust may adhere poorly to filter fibers, and efficiency drops. Filters in such condition should be replaced or reconditioned, as should filters that have reached their terminal (maximum permissible) pressure drop. This category includes both viscous impingement and dry air filters.
2. *Renewable media filters* in which fresh media is introduced into the airstream as needed to maintain essentially constant resistance. This also maintains essentially constant efficiency.
3. *Electronic air cleaners*, which have essentially constant pressure drop and efficiency unless their precipitating elements become severely dust-loaded.

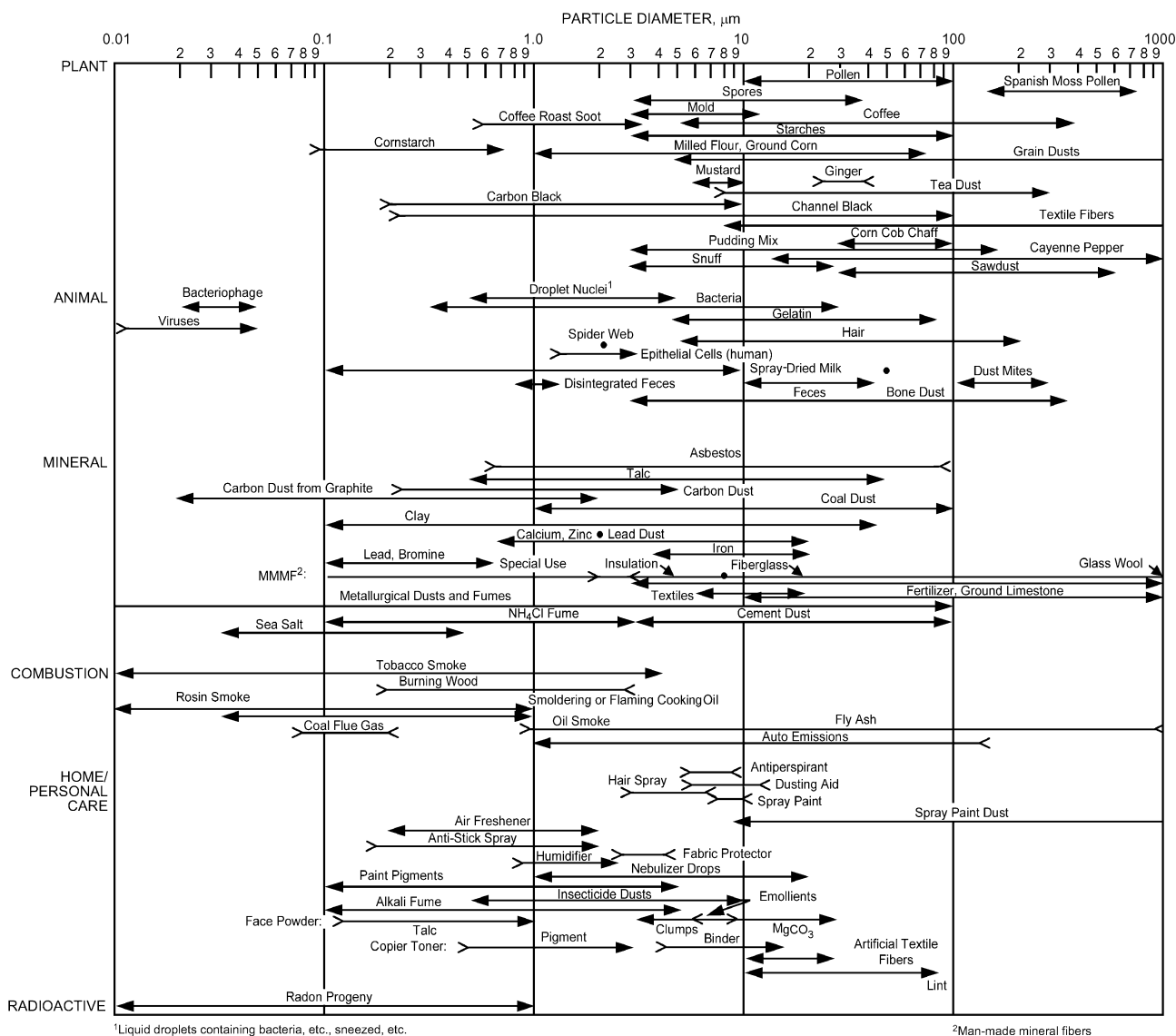


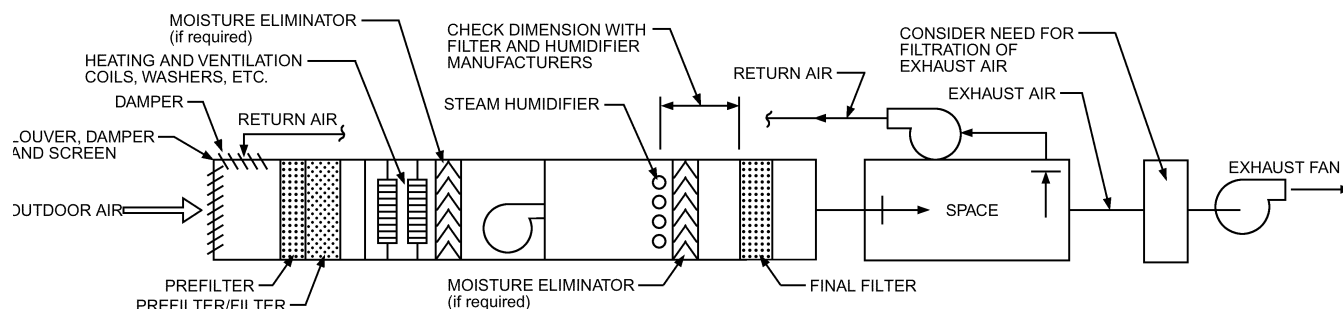
Fig. 18 Sizes of Indoor Particles
 (Figure 3, Chapter 11, 2021 ASHRAE Handbook—Fundamentals) (Owen et al. 1992)

9.2 Fibrous Media Unit Filters

Viscous Impingement Filters. These flat panel filters consist of coarse fibers and have a high porosity. The filter media are coated with a viscous substance, such as oil, which acts as an adhesive for particles that impinge on the fibers. Design air velocity through the media is usually in the range of 250 to 700 fpm (1 to 4 m/s). These filters are characterized by low pressure drop, low cost, and good efficiency on lint, but low efficiency on normal atmospheric dust. They are commonly 1/2 to 4 in. (13 to 100 mm) thick; 1 to 2 in. (25 to 50 mm) nominal thickness is most popular. The thicker configurations have a high dust-holding capacity. Unit panels are available in standard and special sizes up to about 24 by 24 in. (610 by 610 mm). This type of filter is often used as a prefilter to higher efficiency filters.

Many different materials have been used as the filtering medium, including coarse (15 to 60 μm diameter) glass fibers, animal hair, vegetable fibers, synthetic fibers, metallic wools, expanded metals and foils, crimped screens, random-matted wire, and synthetic open-cell foams.

Although viscous impingement filters usually operate in the range of 300 to 600 fpm (1.5 to 3 m/s), they may operate at higher velocities. The limiting factor other than increased flow resistance is the danger of blowing off agglomerates of collected dust and the viscous coating on the filter.



Standard 52.2 MERV	Intended Standard 52.1 Value	Arrestance Value	Example Range of Contaminants Controlled	Example Applications	Sample Air Cleaner Type(s)
HEPA filters					
MERV 20	N/A		0.12 to 0.5 μ m particles: virus (unattached), carbon dust, sea salt, radon progeny, combustion smoke	Cleanroom, pharmaceutical manufacturing and exhaust, radioactive material handling and exhaust, orthopedic and organ transplant surgery, carcinogenic materials, welding fumes	SULPA >99.999% 0.1 to 0.2 μ m IEST type F (ceiling panel)
MERV 19					ULPA >99.999% 0.3 μ m IEST type D (ceiling panel)
MERV 18					HEPA > 99.99% 0.3 μ m IEST type C (ceiling or up to 12 in.deep)
MERV 17					HEPA > 99.97% 0.3 μ m IEST type A (box style 6 to 12 in. deep)
E-1 Range					
MERV 16	Intended to replace 70 to 98% dust-spot efficiency filters	>99%	0.3 to 1.0 μ m size range: bacteria, smoke (ETS), paint pigments, face powder, some virus, droplet nuclei, insecticide dusts, soldering fumes	Day surgery, general surgery, hospital general ventilation, turbo equipment, compressors, welding/soldering air cleaners, prefilters to HEPAs, LEED for existing (EB) and new (NC) commercial buildings, smoking lounges	Box-style wet-laid or lofted fiberglass, box-style synthetic media, minipleated synthetic or fiberglass paper, depths from 4 to 12 in. (), Pocket filters of fiberglass or synthetic media 12 to 36 in).
MERV 15		>99%			
MERV 14		>98%			
MERV 13		>97%			
E-2 Range					
MERV 12	Intended to replace 50 to 80% dust-spot efficiency filters	>97%	1.0 to 3.0 μ m size range: milled flour, lead dust, combustion soot, Legionella, coal dust, some bacteria, process grinding dust	Food processing facilities, air separation plants, commercial buildings, better residential, industrial air cleaning, prefiltration to higher-efficiency filters, schools, gymnasiums	Box-style wet-laid or lofted fiberglass, box-style synthetic media, minipleated synthetic or fiberglass paper, depths from 2 to 12 in. (). Pocket filters either rigid or flexible in synthetic or fiberglass, depths from 12 to 36 in. ().
MERV 11		>95%			
MERV 10		>95%			
MERV 9		>90%			
E-3 Range					
MERV 8	Intended to replace 20 to 60% dust-spot efficiency filters	>90%	3.0 to 10 μ m size range: pollens, earth-origin dust, mold spores, cement dust, powdered milk, snuff, hair spray mist	General HVAC filtration, industrial equipment filtration, commercial property, schools, prefilter to high-efficiency filters, paint booth intakes, electrical/phone equipment protection	Wide range of pleated media, ring panels, cubes, pockets in synthetic or fiberglass, disposable panels, depths from 1 to 24 in. ().
MERV 7		>90%			
MERV 6		>85%			
MERV 5		>85%			
MERV 4	<20%	>70%	Arrestance method	Protection from blowing large particle dirt and debris, industrial environment ventilation air	Inertial separators
MERV 3	<20%	>70%			
MERV 2	<20%	>65%			
MERV 1	<20%	<65%			

Note: MERV for non-HEPA/ULPA filters also includes test airflow rate, but it is not shown here because it is of no significance for the purposes of this table.
N/A = not applicable.

Fig. 19 Typical Filter Locations and Cross Reference and Application Guidelines
(Figure 3 and Table 2, Chapter 29, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

The rate of filter loading depends on the type and concentration of the dirt being handled and on the operating cycle of the system. Manometers or draft gages are often installed to measure the pressure drop across the filter bank and thereby indicate when the filter needs servicing. The final allowable pressure drop may vary from one type of filter to another. The decline in filter efficiency that occurs when all the viscous coating has been absorbed by the collected dust, rather than the increased resistance due to dust load, may be the limiting factor in operating life.

Dry Air Filters. The media used in dry air filters are random fiber mats or blankets of varying thicknesses, fiber sizes, and densities. Media of bonded glass fiber, cellulose fibers, wool felt, synthetics, and other materials have been used commercially. The medium in filters of this class is frequently supported by a wire frame in the form of pockets or V-shaped pleats. In other designs, the media may be self-supporting because of inherent rigidity or because airflow inflates it into extended form. Pleating of the media provides a high ratio of media area to face area, thus allowing reasonable pressure drop despite the density and fineness of the media.

The efficiency of dry air filters is usually higher than that of viscous impingement filters, and the variety of media available makes it possible to furnish almost any degree of cleaning efficiency desired. Most dry filter media and filter configurations also have higher dust-holding capacities than viscous impingement filters. Coarse prefilters placed ahead of high-efficiency dry filters may be economically justified by the even longer life they give the main filters.

Electronic Air Cleaners. These ionizing filters are efficient, low pressure drop devices for removing fine dust and smoke particles. Collector plates are often coated with a special oil as an adhesive. Cleaning is generally accomplished by washing the cells in place with hot water from a water hose or by a fixed or moving nozzle system.

Electrical forces drive most particles to the collecting surface but cannot hold them there. In fact, after a particle touches the collecting surface, the electrical force reverses and tends to pull it off, and the dust is held only by intermolecular adhesion forces. It is, therefore, very important with the washed type of electronic air cleaner to ensure that either the dust is naturally adherent or that the plates are always covered with adhesive.

Electronic air cleaners, however, are often used without any adhesive treatment on the plates. Under such conditions, the precipitator forms agglomerates that eventually blow off the plates. They must be followed downstream by a secondary filter or storage section. The dry agglomerates produced in the precipitator are allowed to blow off and be caught by the downstream filter. An automatic replaceable media filter for catching these agglomerates gives a combination with a high degree of cleaning efficiency and also the convenient maintenance associated with an automatic filter.

10 Air-to-Air Energy Recovery Equipment

Energy can be recovered from exhaust air, as well as transferred from one location to another. This can be done using rotary devices, heat pipe heat exchangers, coil heat recovery loops, twin tower heat recovery loops, fixed plate exchangers, and thermosiphon heat exchangers.

Table 2 provides comparative data for common types of air-to-air energy recovery devices. The following sections describe the construction, operation, and unique features of the various devices.

10.1 Performance Rating of Equipment

Performance of air-to-air heat exchangers is usually expressed in terms of their effectiveness in transferring (1) sensible energy (dry-bulb temperature), (2) latent energy (humidity ratio), or (3) total energy (enthalpy). The effectiveness ε of a heat exchanger is defined as follows:

$$\varepsilon = \frac{\text{Actual transfer for given device}}{\text{Maximum possible transfer between airstreams}}$$

Referring to Figure 20,

$$\varepsilon = \frac{W_e(X_1 - X_2)}{W_{\min}(X_1 - X_3)} = \frac{W_s(X_4 - X_3)}{W_{\min}(X_1 - X_3)} \quad (6)$$

Table 2 Comparison of Air-to-Air Energy Recovery Devices (I-P)
(Table 3, Chapter 26, 2016 *ASHRAE Handbook—HVAC Systems and Equipment*)

	Fixed Plate	Membrane Plate	Energy Wheel	Heat Wheel	Heat Pipe	Runaround Coil Loop	Thermosiphon	Liquid Desiccant
Airflow arrangements	Counterflow Cross flow ^a	Counterflow Cross flow ^a	Counterflow Parallel flow	Counterflow	Counterflow Parallel flow	—	Counterflow Parallel flow	—
Equipment size range, cfm (L/s)	50 and up (25 and up)	50 and up (25 and up)	50 to 74,000 and up (25 to 35 000 and up)	50 to 74,000 and up (25 to 35 000 and up)	100 and up (50 and up)	100 and up (50 and up)	100 and up (50 and up)	—
Typical sensible effectiveness ($m_s = m_e$), % ^c	50 to 75	55 to 75	65 to 80	65 to 80	40 to 60 ^b	45 to 65 ^b	40 to 60	40 to 60 ^b
Typical latent effectiveness, % ^c	0	25 to 60	50 to 80	0	0	0	0	50 to 75 ^{b,d}
Total effectiveness, % ^c	20 to 50	35 to 70	55 to 80	25 to 60	15 to 35	—	—	40 to 75 ^d
Face velocity, fpm (m/s)	200 to 1000 (1 to 5)	200 to 600 (1 to 3)	500 to 1000 (2.5 to 5)	400 to 1000 (2 to 5)	400 to 800 (2 to 4)	300 to 600 (1.5 to 3)	400 to 800 (2 to 4)	300 to 450 (1.5 to 2.2)
Pressure drop, in. of water (Pa)	0.4 to 4 (100 to 1000)	0.4 to 2 (100 to 500)	0.4 to 1.2 (100 to 300)	0.4 to 1.2 (100 to 300)	0.6 to 2 (150 to 500)	0.6 to 2 (150 to 500)	0.6 to 2 (150 to 500)	0.7 to 1.2 (170 to 300)
EATR, %	0 to 2	0 to 5	0.5 to 10	0.5 to 10	0 to 1	0	0	0
OACF	0.97 to 1.06	0.97 to 1.06	0.99 to 1.1	1 to 1.2	0.99 to 1.01	1.0	1.0	1.0
Temperature range, °F (°C)	−75 to 1470 (−60 to 800)	−40 to 140 (−40 to 60)	−65 to 1470 (−55 to 800)	−65 to 1470 (−55 to 800)	−40 to 200 (−40 to 93)	−50 to 930 (−45 to 500)	−40 to 104 (−40 to 40)	−40 to 115 (−40 to 46)
Typical mode of purchase	Exchanger only Exchanger in case Exchanger and blowers Complete system	Exchanger only Exchanger in case Exchanger and blowers Complete system	Exchanger only Exchanger in case Exchanger and blowers Complete system	Exchanger only Exchanger in case Exchanger and blowers Complete system	Exchanger only Exchanger in case Exchanger and blowers Complete system	Coil only Complete system	Exchanger only Exchanger in case	Complete system
Advantages	No moving parts Low pressure drop Easily cleaned	No moving parts Low pressure drop Low air leakage Moisture/mass transfer	Moisture/mass transfer Compact large sizes Low pressure drop Easily cleaned Available on all ventilation system platforms	Compact large sizes Low pressure drop Easily cleaned	No moving parts except tilt Fan location not critical Allowable pressure differential up to 2 psi (15 kPa)	Exhaust airstream can be separated from supply air Fan location not critical	No moving parts Exhaust airstream can be separated from supply air Fan location not critical	Latent transfer from remote airstreams Efficient micro-biological cleaning of both supply and exhaust airstreams
Limitations	Large size at higher flow rates	Few suppliers Long-term maintenance and performance unknown	Supply air may require some further cooling or heating Some EATR without purge	Some EATR with purge	Effectiveness limited by pressure drop and cost Few suppliers	Predicting performance requires accurate simulation model	Effectiveness may be limited by pressure drop and cost Few suppliers	Few suppliers Maintenance and performance unknown
Heat rate control (HRC) methods	Bypass dampers and ducting	Bypass dampers and ducting	Bypass dampers and wheel speed control	Bypass dampers and wheel speed control	Tilt angle down to 10% of maximum heat rate	Bypass valve or pump speed control	Control valve over full range	Control valve or pump speed control over full range

^aRated effectiveness values are for balanced flow conditions for cross flow. Effectiveness values increase slightly if flow rates of either or both airstreams are higher than flow rates at which testing is done.

^bData not based on third-party certified data.

^cData based on typical range of third-party certified data.

^dFace velocity of 250 to 500 fpm (1.27 to 2.54 m/s).

EATR = exhaust air transfer ratio

OACF = outdoor air correction factor

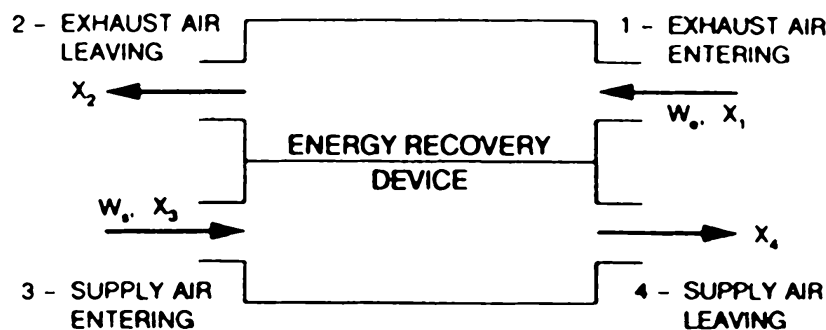


Fig. 20 Nomenclature for Effectiveness Evaluation

where

ε = sensible, latent, or total heat effectiveness

X = dry-bulb temperature, humidity ratio, or enthalpy at locations indicated in Figure 20

For the latent and total heat effectiveness,

W_s = mass flow rate of supply

W_e = mass flow rate of exhaust

For the sensible heat effectiveness,

W_s = (specific heat) (mass flow rate) for the supply

W_e = (specific heat) (mass flow rate) for the exhaust

W_{min} = smaller of W_s and W_e

The leaving supply air condition is then:

$$X_2 = X_1 - \varepsilon(X_1 - X_3)(W_{min}/W_s) \quad (7)$$

The leaving exhaust air condition is:

$$X_4 = X_3 + \varepsilon(X_1 - X_3)(W_{min}/W_e) \quad (8)$$

The effectiveness of a particular air-to-air energy recovery device is a function of several variables, including the supply and exhaust mass flow rates and the energy transfer characteristics of the device. Because of this combination, performance data must be established for each device.

10.2 Economics of Air-to-Air Energy Recovery

In analyzing the advisability of any air-to-air energy recovery application, one must usually consider both first and operating costs.

First Cost

- Energy recovery device
- Installing energy recovery device
- Additional cost of building space to accommodate apparatus
- Additional ductwork to accommodate device or cost of piping for liquid energy transfer
- Larger fans and/or motors to overcome air pressure loss of energy recovery device
- Additional air filtration required (if any)
- Capacity controls
- Any auxiliary heaters required for frost control
- Savings of boiler or heating equipment due to reduced design load
- Savings of heating coils and associated piping and pumps due to reduced design capacity
- Savings of chiller or cooling plant due to reduced design load
- Savings of cooling coils and associated piping and pumps due to reduced design capacity
- Savings of electric power requirement

Operating Cost

- Maintaining the energy recovery device
- Operating fans to overcome additional static pressure
- Maintaining additional filtration (if any)
- Operating energy recovery device drive, pumps, controls, and defrost heaters
- Savings of annual heating energy, based on weather data for the system location
- Savings of annual cooling energy, based on weather data for the system location

10.3 Rotary Air-to-Air Energy Exchangers

A rotary air-to-air energy exchanger, often called a *heat wheel*, is a revolving cylinder filled with an air-permeable medium with a large internal surface area for contact with the air passing through it. Adjacent supply and exhaust airstreams each flow through half the exchanger in a counterflow pattern (Figure 21). Media material may be selected to recover either sensible heat only or total heat (sensible heat plus latent heat). With total heat recovery, the unit is called an *energy wheel*.

Sensible heat is recovered (transferred) as the medium picks up and stores heat from the hot airstream and gives it up to the cold one. Latent heat is transferred as the medium (1) condenses moisture from the airstream having the higher humidity ratio (by means of absorption for liquid desiccants and adsorption for solid desiccants) with a simultaneous release of heat and (2) then releases the moisture through evaporation (and heat pickup) into the airstream with the lower humidity ratio. Thus, more moist air is dried while drier air is humidified. In total heat transfer, both sensible and latent heat recovery processes take place simultaneously.

Choice of structural material for the casing, rotor structure, and medium of a rotary energy exchanger is influenced by the contaminants, dew point, and temperature of the exhaust air, as well as by the properties of the supply air. Aluminum and steel are the usual casing and rotor materials for normal comfort ventilating systems. Exchanger media are fabricated from metal, mineral, or man-made materials and classified as providing either random flow or directionally oriented flow through their structures.

The performance of a rotary energy exchanger is defined by the exchanger's effectiveness and the media pressure drops. Face velocities for most energy recovery applications range from 500 to 800 fpm (2.5 to 4.0 m/s). Low face velocities give lower pressure drop, higher effectiveness, and lower operating costs, but require larger size units with higher capital costs and more installation space.

Typical pressure drops for various types of media at 500 fpm (2.5 m/s) vary from 0.4 to 0.7 in. (100 to 170 kPa) of water. (Consult the manufacturer's catalog in each case for actual data.) Average effectiveness values for sensible and total heat exchangers lie in the 70% to 85% range for equal supply and exhaust air mass flow rates and usual exchanger face velocities.

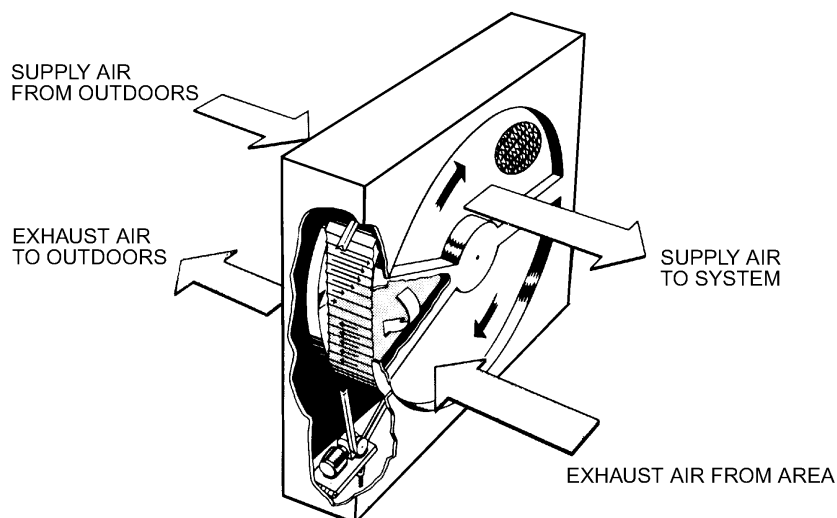


Fig. 21 Rotary Energy Wheel

(Figure 7, Chapter 26, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

Rotary energy wheels are available in single units to 68,000 cfm (32 m³/s) capacity; they are usually not larger than 14 ft (4.25 m) in diameter due to difficulty in shipping, erecting, and fitting into buildings. Multiple units may be used to provide greater single-system capacities. Units are available for temperatures from –70°F to 1500°F (–60°C to 800°C). When installed horizontally (vertical airflow), units greater than 8 ft (2.4 m) in diameter may require special structural considerations due to their size and weight.

10.4 Coil Energy Recovery Loops

The coil energy recovery or “runaround loop” system uses extended-surface, finned-tube water coils placed in the supply and exhaust airstreams of a building or process. The coils are connected in a closed loop via counterflow piping and an intermediate heat transfer fluid of water (typically) or a freeze-protected solution pumped through the coils.

This system allows energy to transfer from the warmer to the cooler airstream. In a typical comfort-to-comfort application, the system is seasonally reversible—the supply air preheats when outdoor air is cooler than the exhaust air and precools when the outdoor air is warmer. This system operates generally for sensible heat recovery only.

As with other air-to-air energy recovery equipment, measures must be taken to prevent potential freezing of exhaust air condensate. A dual-purpose, three-way temperature control valve is used to prevent exhaust coil frosting. The valve is controlled to maintain an entering solution temperature to the exhaust coil of not less than 30°F (–1°C). This is accomplished by bypassing some of the warmer solution from the supply air coil. The valve can also ensure that a prescribed leaving air temperature from the supply air coil is not exceeded for those applications where the energy recovered must be limited.

This system affords a high degree of flexibility, which makes it well-suited for renovation and industrial applications. The system accommodates remotely located supply and exhaust ducts. It also allows simultaneous energy recovery from multiple supply and exhausts. A basic arrangement of the coil energy recovery loop is depicted in Figure 22.

10.5 Heat Pipe Heat Exchangers

A heat pipe heat exchanger has no moving parts; it is a passive energy recovery device. Although it appears similar to a standard steam or chilled water coil, it differs in two major aspects. As shown in Figure 23, each tube is an individual heat pipe, that operates independently and acts as a superconductor of heat. Secondly, the heat pipe heat exchanger is divided into two airflow paths. Hot air passes through one side of the exchanger and cold air through the other side in the opposite direction, that is, in a counterflow arrangement. Sensible energy from the hot air is transferred by the heat pipes to the other side of the exchanger, where

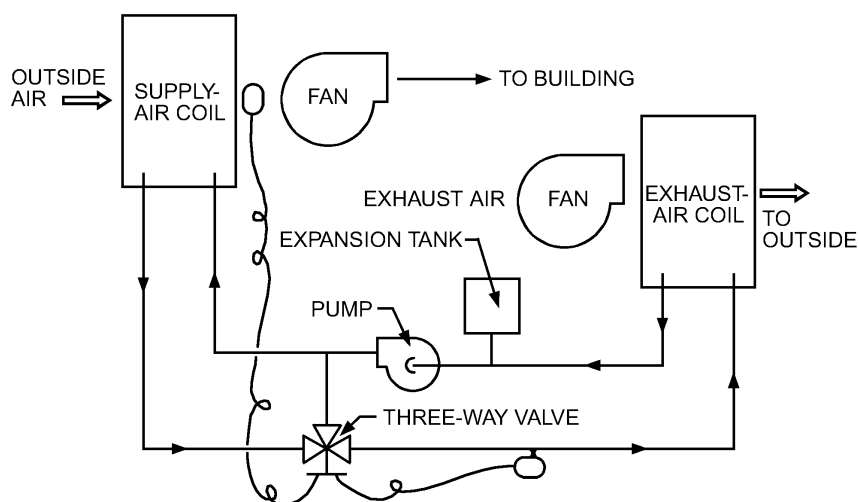


Fig. 22 Arrangement of Coil Energy Recovery Loop
(Figure 14, Chapter 26, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

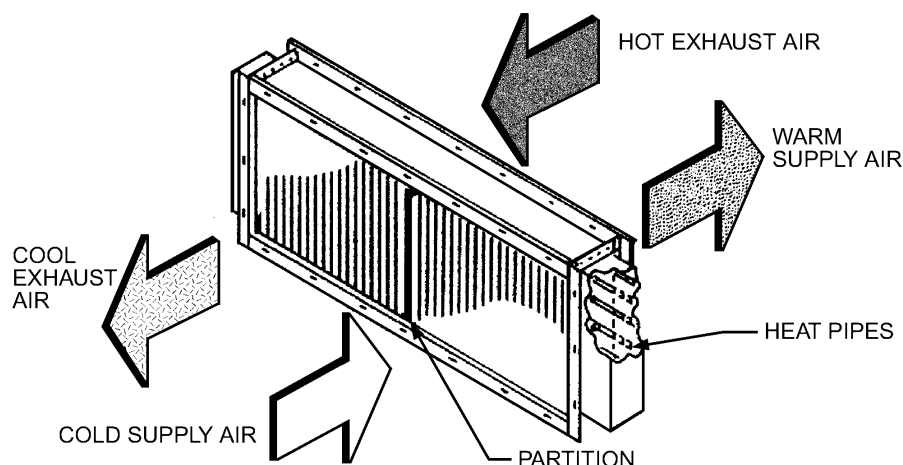


Fig. 23 Heat Pipe Assembly
(Figure 20, Chapter 26, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

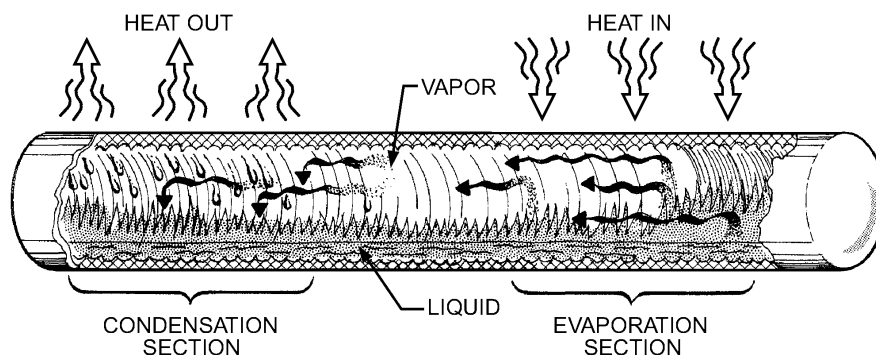


Fig. 24 Heat Pipe
(Figure 21, Chapter 26, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

it is captured by the cold air, thereby warming it. Although the heat pipes span the width of the unit, a sealed partition separates the two airstreams, preventing any cross-contamination between them.

Figure 24 is a schematic of a heat pipe. A heat pipe is a tube that is fabricated with a capillary wick structure, filled with a refrigerant or suitable change-of-phase heat transfer fluid, and permanently sealed.

Basically, the heat pipe operates on a condensation/evaporation cycle that is continuous as long as there is a temperature difference to drive the process. Energy transfer within a heat pipe is accomplished with a very small temperature drop; a heat pipe is essentially an isothermal device.

Thermal energy applied to either end of the pipe causes the fluid at the end to vaporize. The vapor then travels to the other end of the pipe where the removal of thermal energy causes the vapor to condense into liquid again, thus giving up the latent heat of condensation. The condensed liquid then flows back to the evaporator section (i.e., the hot end) to be reused, completing the cycle.

Heat pipes have a finite heat transfer capacity affected by such factors as wick design, tube diameter, working fluid, and tube orientation relative to horizontal. Design face velocities for heat pipe heat exchangers range from 400 to 800 fpm (2.0 to 4.1 m/s), with 450 to 550 fpm (2.3 to 2.8 m/s) most common. Design face velocities are generally established on allowable pressure drop rather than recovery performance.

Pressure drops at 60% effectiveness range from 0.4 to 0.7 in. of water at (100 to 170 Pa) 400 fpm (2.0 m/s) up to 1.5 to 2.0 in. of water (370 to 500 Pa) at 800 fpm (4.1 m/s). Recovery performance, or effectiveness, decreases with increasing velocity, but the effect is not as pronounced as with pressure drop.

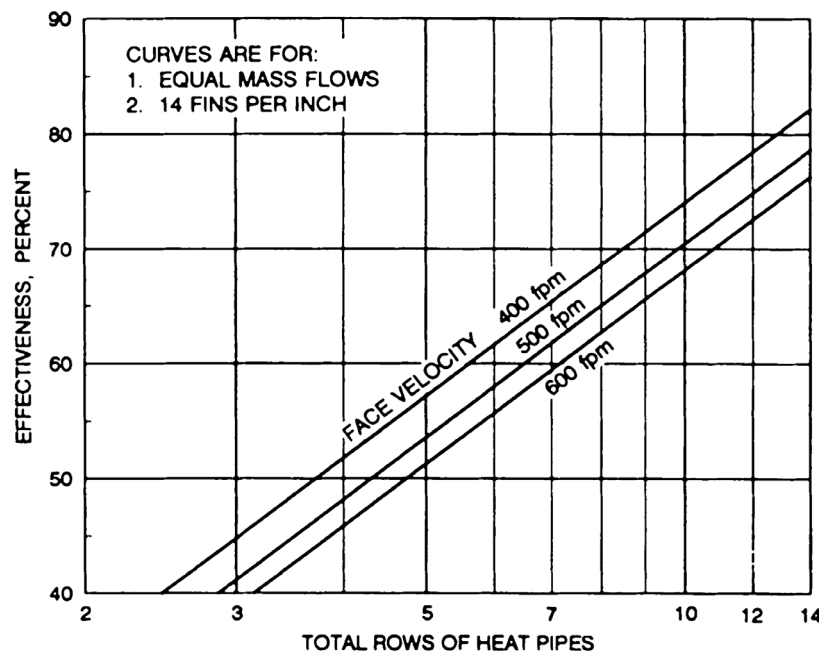


Fig. 25 Heat Pipe Exchanger Effectiveness

(Figure 22, Chapter 26, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

Available fin designs include continuous corrugated plate fins, continuous flat plate fins, and spiral fins. These fin designs and tube spacing cause the variation in pressure drop, noted previously, at a given face velocity.

Figure 25 presents a typical effectiveness curve for various face velocities and rows of tubes. As the number of rows increases, effectiveness increases at a decreasing rate. For example, doubling the rows of tubes in a 60% effective heat exchanger increases the effectiveness to 75%.

10.6 Fixed Plate Exchangers

The plate heat exchanger is a static device that has no leakage between airstreams. Since it uses no secondary heat transfer medium, such as water or refrigerant, its temperature range is the broadest of all air-to-air energy recovery equipment.

A fixed surface plate exchanger can be classified as (1) a **pure-plate heat exchanger**, consisting of only a primary heat transfer surface, or (2) a **plate-fin heat exchanger**, which is made up of alternate layers of separate plates and interconnecting fins. The pure-plate exchanger is usually a counterflow design, whereas the basic plate-fin exchanger is a crossflow design with combinations sometimes arranged to approach a counterflow unit. Counterflow provides the greatest temperature difference for maximum heat transfer, but crossflow can sometimes give more convenient air connections.

Fixed surface plate exchangers have no moving parts. Alternate layers of plates, separated and sealed (referred to as the heat exchanger core), form the exhaust and supply airstream passages (Figure 26). Plate spacings range from 0.1 to 0.5 in. (2.5 to 13 mm), depending on the design and application. Heat transfers directly from the warm airstreams through the separating plates into the cool airstreams.

Normally, both latent heat of condensation, from moisture condensed as the temperature of the warm (exhaust) airstream drops below its dew point, and sensible heat are conducted through the separating plates into the cool (supply) airstream. Thus, latent energy but not actual moisture may be transferred. Recovering upward of 80% of the available waste exhaust heat is not uncommon.

Fixed-plate heat exchangers can be made from permeable microporous membranes designed to maximize moisture and energy transfer between airstreams while minimizing air transfer. Suitable permeable microporous membranes for this emerging technology include cellulose, polymers, and other synthetic materials such as hydrophilic electrolyte. Hydrophilic electrolytes are made from sulphonation chemistry techniques

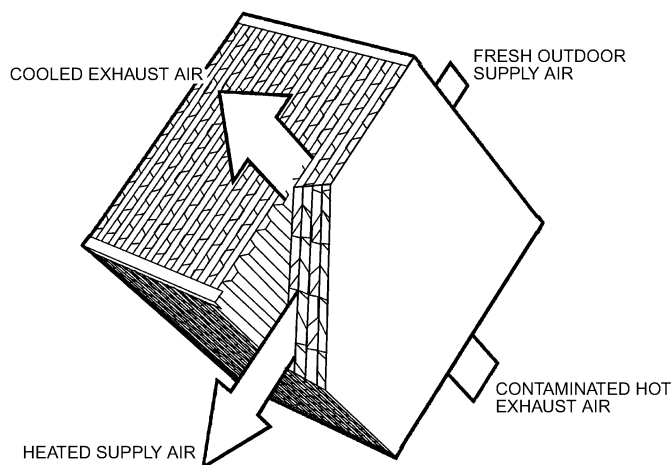


Fig. 26 Fixed Plate Cross-Flow Heat Exchanger
(Figure 4, Chapter 26, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

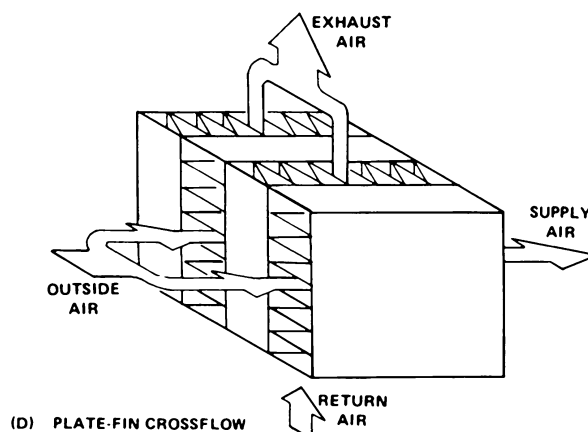
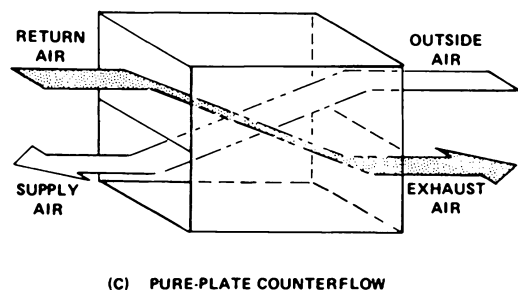
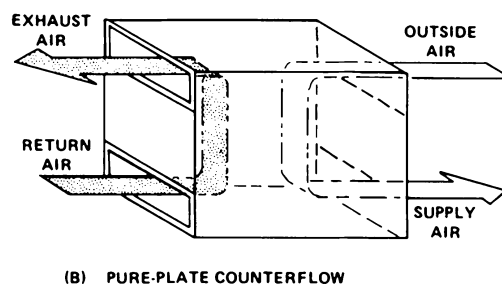
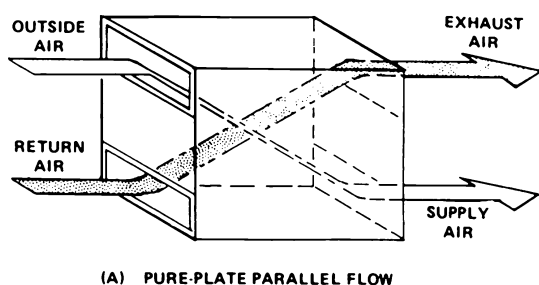


Fig. 27 Pure-Plate and Plate-Fin Models

and contain charged ions that attract polar water molecules; adsorption and desorption of water occur in vapor state.

Plate exchangers are of many proprietary designs, with weights, sizes, and flow patterns depending on the manufacturer (Figure 27). Most manufacturers of plate exchangers offer the equipment in modular design.

10.7 Thermosiphon Heat Exchangers

Thermosiphon heat exchangers use the natural gravity circulation of a boiling and condensing intermediate fluid to transfer energy between exhaust and supply airstreams. They may be classified as sealed tube thermosiphons and coil loop thermosiphons. These two types are illustrated in Figures 28 and 29.

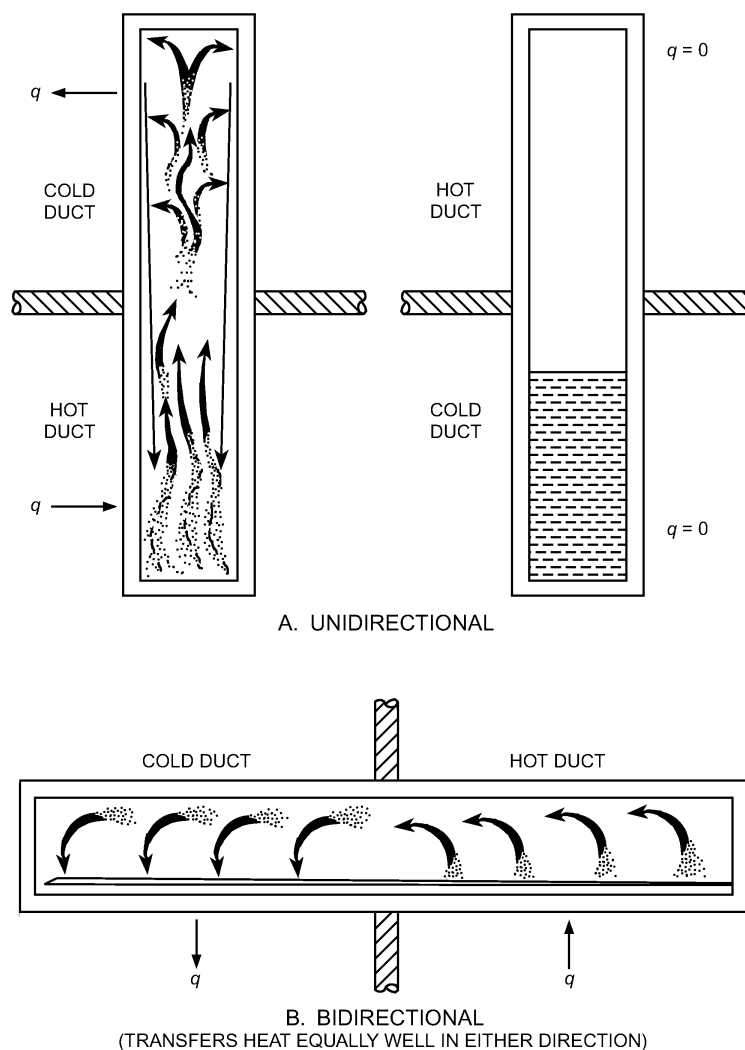


Fig. 28 Sealed-Tube Thermosiphons

The sealed tube type is similar to a heat pipe and is often given that name. The only distinction made between the two is that heat pipes are usually considered to use, if not solely rely on, capillary forces to cause the intermediate liquid to flow from the cold to the hot end of the tubes, whereas the thermosiphon tubes rely only on gravity. The coil loop type is similar in appearance to the coil energy recovery loop discussed previously. The most obvious difference is the absence of a circulating pump in the thermosiphon loop and the need for evaporator and condenser coils rather than single-phase liquid coils.

Example 2 Determine the leaving conditions for an energy wheel exchanger with the following conditions (see Figure 30):

Cooling

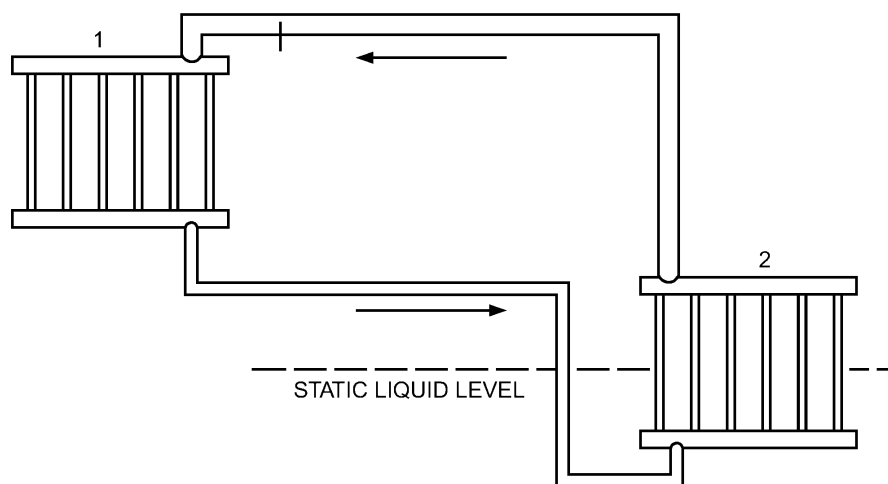
1. Design conditions: outdoor air = 94°F (34.4°C) DB, 77°F (25°C) WB
2. Design conditions: space = 75°F (23.9°C) DB, 62.5°F (16.9°C) WB

Heating

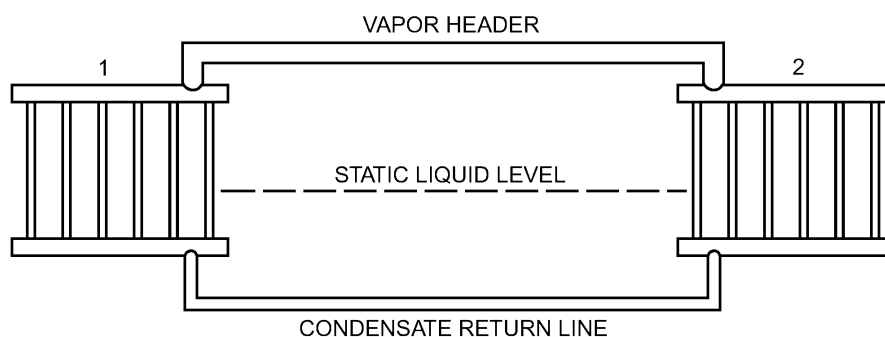
1. Design conditions: outdoor air = 0°F (−17.8°C) DB, −3°F (−19.4°C) WB
2. Design conditions: space = 75°F (23.9°C) DB, 59.5°F (15.3°C) WB

Cooling and Heating

Exchanger effectiveness = 80% on sensible heat and 80% on latent heat. Equal mass flow rates.



A. UNIDIRECTIONAL LOOP
(TRANSFERS ENERGY ONLY FROM 2 TO 1)



B. BIDIRECTIONAL LOOP
(TRANSFERS ENERGY IN EITHER DIRECTION)

Fig. 29 Coil-Type Thermosiphon Loops

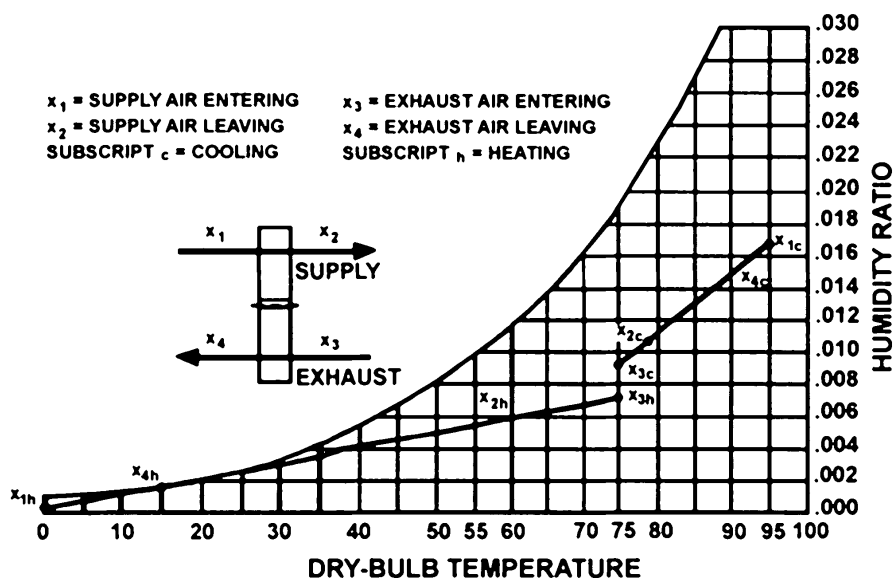


Fig. 30 Conditions for Example 2

Solution

Dry-bulb temperature and humidity ratio are calculated from Equation (8).

Summer dry-bulb temperature

$$X_2 = X_1 - \varepsilon(X_1 - X_2) = 94 - 0.8(94 - 75) = 78.8^\circ\text{F} (26.0^\circ\text{C})$$

Summer humidity ratio

$$X_2 = X_1 - \varepsilon(X_1 - X_2) = 0.0162 - 0.8(0.0162 - 0.0092) = 0.0106 \text{ lb/lb} (0.0106 \text{ kg/kg})$$

Winter dry-bulb temperature

$$X_2 = X_1 - \varepsilon(X_1 - X_2) = 0 - 0.8(0 - 75) = 60^\circ\text{F DB} (15.5^\circ\text{C})$$

Winter humidity ratio

$$X_2 = X_1 - \varepsilon(X_1 - X_2) = 0.0001 - 0.8(0.0001 - 0.0074) = 0.0059 \text{ lb/lb} (0.0059 \text{ kg/kg})$$

Example 3 Determine the leaving conditions for an energy wheel exchanger with the following comfort conditions:

Cooling

1. Design conditions. Outdoor air = 95°F (35°C) DB, 78°F (25.6°C) WB, and 41.3 Btu/lb (96.1 kJ/kg).
2. Design conditions. Space = 75°F (23.9°C) DB, 62.5°F (17°C) WB, and 28.3 Btu/lb (65.8 kJ/kg) of dry air.

Heating

1. Design conditions. Outdoor air = 20°F (−6.7°C) DB and 5.0 Btu/lb (11.6 kJ/kg) of dry air.
2. Design conditions. Space = 75°F (23.9°C) DB and 22.4 Btu/lb (52.1 kJ/kg) of dry air.

Cooling and Heating

Exchanger effectiveness = 80% on sensible heat and 65% on total heat. Equal mass flow rates.

Solution

Dry-bulb temperature and enthalpy are calculated from Equation (8).

Summer dry-bulb temperature

$$X_2 = X_1 - \varepsilon(X_1 - X_3) = 95 - 0.8(95 - 75) = 79^\circ\text{F} (26.1^\circ\text{C})$$

Summer enthalpy

$$X_2 = X_1 - \varepsilon(X_1 - X_3) = 41.3 - 0.65(41.3 - 28.3) = 32.9 \text{ Btu/lb} (76.5 \text{ kJ/kg})$$

Winter dry-bulb temperature

$$X_2 = X_1 - \varepsilon(X_1 - X_3) = 20 - 0.8(20 - 75) = 64^\circ\text{F} (17.8^\circ\text{C})$$

Heating enthalpy

$$X_2 = X_1 - \varepsilon(X_1 - X_3) = 0.5 - 0.65(5.0 - 22.4) = 16.3 \text{ Btu/lb} (37.9 \text{ kJ/kg})$$

Example 4 Calculate recovered temperatures and volumes (measured at point 2), as well as energy savings, for a heat wheel exchanger handling 8000 cfm (3.8 m³/s) of process exhaust at 300°F (149°C) measured at point 3, considered dry air. Assume an equal mass of makeup air at winter design conditions of 0°F (−17.8°C). Exchanger effectiveness is 80% at equal mass flows. Neglect cross leakage and purge volume. (See Figure 31.)

Solution

1. For conditions at 300°F (149°C)

$$8000 \text{ ft}^3/\text{min} / 19.13 \text{ ft}^3/\text{lb} = 418 \text{ lb/min} (189.3 \text{ kg/min})$$

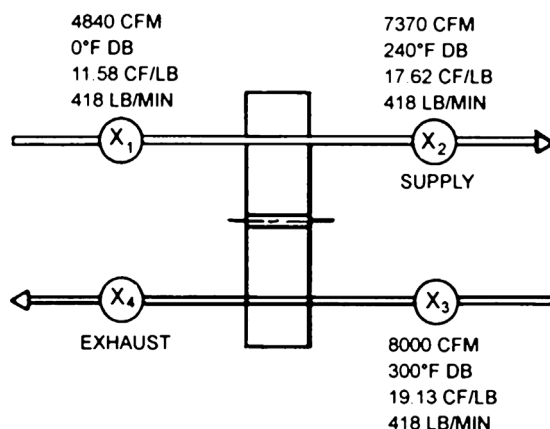


Fig. 31 Schematic and Conditions for Example 4

2. For conditions of 0°F (−17.8°C)

$$v_1 = 11.58 \text{ ft}^3/\text{lb}$$

$$418 \text{ lb/min} \times 11.58 \text{ ft}^3/\text{lb} = 4840 \text{ cfm at Point 1 (2.28 m}^3/\text{s)}$$

3. Temperature at Point 2 from Equation (8):

$$X_2 = X_1 - \varepsilon(X_1 - X_3) = 0 - 0.8(0 - 300) = 240^\circ\text{F DB (115.5}^\circ\text{C)}$$

4. Specific volume at 240°F DB (115.5°C)

$$v_2 = 17.62 \text{ ft}^3/\text{lb}$$

$$418 \text{ lb/min} \times 17.62 \text{ ft}^3/\text{lb} = 7370 \text{ cfm (3.5 m}^3/\text{s)}$$

5. Savings at winter design conditions [specific heat = 0.242 Btu/(lb·°F)]

$$418 \text{ lb/min} \times 60 \text{ min/h} \times 0.242 \text{ Btu/(lb} \cdot ^\circ\text{F)} \times (240 - 0) = 1,460,000 \text{ Btu/h (428 kW)}$$

Example 5 Determine the leaving conditions for a heat pipe exchanger with the following comfort conditions (see Figure 32).

Cooling

1. Design conditions, outdoor air = 95°F DB (35°C)
2. Design conditions, space = 75°F DB (23.9°C)

Heating

1. Design conditions, outdoors = 10°F DB (−12.2°C)
2. Design conditions, space = 75°F DB (23.9°C)

Cooling and Heating

Exchanger effectiveness = 65% on sensible heat at equal mass flows.

Solution

Dry-bulb temperature is calculated from Equation (8).

Cooling dry-bulb temperature

$$X_2 = X_1 - \varepsilon(X_1 - X_3) = 95 - 0.65(95 - 75) = 82^\circ\text{F (27.8}^\circ\text{C)}$$

Heating dry-bulb temperature

$$X_2 = X_1 - \varepsilon(X_1 - X_3) = 10 - 0.65(10 - 75) = 52.3^\circ\text{F (11.3}^\circ\text{C)}$$

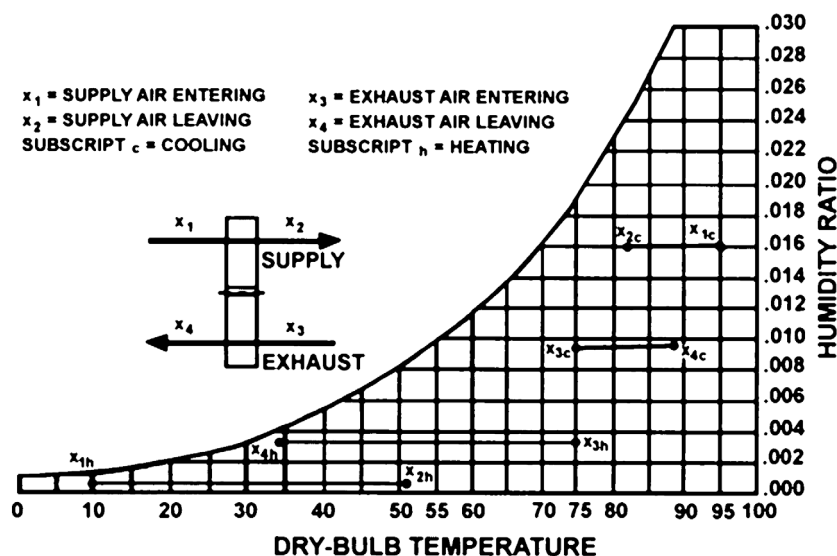


Fig. 32 Conditions for Example 5

11 Economizers

An economizer uses outdoor air to reduce the refrigeration required to provide cooling for the building when the outdoor air dry-bulb temperature is low enough to provide for the sensible cooling needs or reduce the refrigeration requirements to do so. Either the air-side or the water-side economizer is an attractive option for reducing energy costs with self-contained HVAC systems. Although climate often dictates which economizer type is selected, either one can provide advantages.

The air-side economizer includes an outdoor air damper, relief damper, return air damper, filters, actuator, and linkage. Economizer controls are usually a factory-installed option. The air-side economizer takes advantage of cool outdoor air to either assist mechanical cooling or, if the outdoor air is sufficiently cool, provide total system cooling. However, if the building has significant simultaneous heating and cooling requirements, the interaction of the economizer with an installed heat recovery system must be thoroughly analyzed.

Self-contained units usually do not include return air fans. It is necessary to include a variable-volume relief fan unit when air-side economizers are employed. The relief fan volume is generally controlled with discharge dampers in response to building space pressure. The relief fan is off and discharge dampers are closed when the air-side economizer is inactive. However, the installed cost of an air-side economizer is generally higher than that for a water-side economizer.

Typically, in an air-side economizer, an enthalpy sensor or dry-bulb temperature probe energizes the unit to bring in outdoor air as the first stage of cooling. An outdoor air damper modulates the flow to meet a design temperature, and when outdoor air can no longer provide enough cooling, the refrigeration system is energized.

The water-side economizer (see Figure 33) consists of a water coil located upstream of the main cooling coil. All economizer control valves, piping between the economizer coil and the condenser, and economizer control wiring can be factory installed. The water-side economizer takes advantage of low cooling tower water temperature (approaching ambient wet-bulb temperature) to either precool the entering air, assist mechanical cooling, or, if the cooling water is cold enough, to provide total system cooling. If the economizer is unable to maintain the supply air set point for VAV units or zone set point for constant-volume units, factory-mounted controls integrate economizer and compressor operation to meet cooling requirements.

Cooling water flow rate is controlled by two valves (Figure 30), one at the economizer coil inlet (A) and one in the bypass loop to the condenser (B). Two control methods are common—constant water flow and variable water flow.

Standard modulating control allows constant condenser water flow during operation. The two control valves are wired for complementary operation, where one valve is drive open while the other is drive closed.

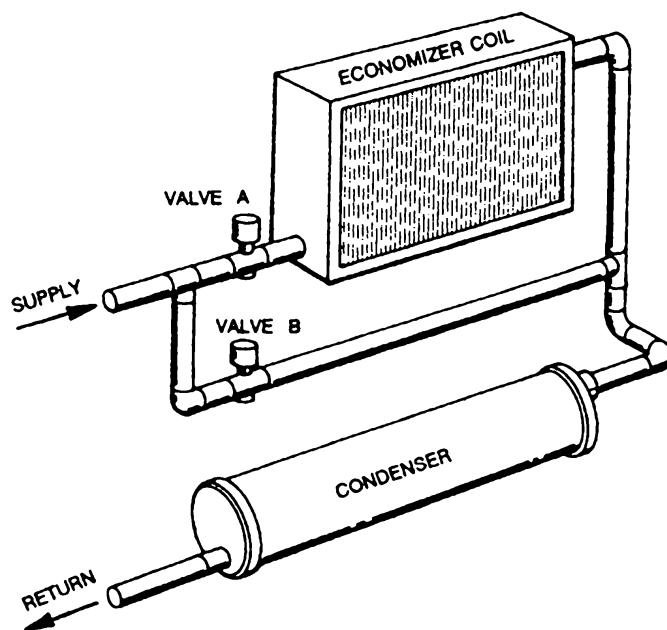


Fig. 33 Water-Side Economizer

Energy-saving modulating valve control allows variable condenser water flow during operation. The valve in the bypass loop (B) is an on-off valve and is closed when the economizer is enabled. Water flow through the economizer coil is modulated by valve A. As the cooling load increases, valve A opens, increasing water flow through the economizer coil. If the economizer is unable to completely satisfy the cooling requirements, the control system integrate economizer and compressor operation. When the system is not in the cooling mode, both valves are closed.

12 References

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13 Problems

Problem 1 Air enters a coil at 95°F DB and 78°F WB temperature and leaves at 62°F DB and 60°F WB temperature. The condensate is assumed to be at a temperature of 56°F. Find the total, latent, and sensible cooling loads on the coil with air at 14.7 psia.

Problem 2 Air enters a direct expansion coil at 85°F (29.4°C) DB and 70°F (21.1°C) WB and leaves at 62°F (16.7°C) DB and 90% rh.

1. How much sensible heat and how much latent heat is removed from the air by the coil?
2. How much condensate drains off the coil?

Problem 3 Air enters a direct-expansion coil at 90°F (32.2°C) DB 60% rh, and leaves the coil at 60°F (15.6°C) DB, 95% rh. Find the following:

1. Heat removed from air
2. Moisture condensed from air
3. SHR for the condition line

Problem 4 Water flowing at 60 lb/min and at 51°F is chilled in an evaporator to 40°F. The heat transfer area is 20 ft² and the heat exchanger has an overall heat transfer coefficient of 60 Btu/h·ft²·°F. The direct-expansion evaporator uses R-12 and operates at 35°F. Find the evaporator effectiveness.

Problem 5 Outdoor air at 35°F and 70% rh is supplied to an air-conditioning apparatus. Recirculated air is returned from the plant at 69°F DB and 40% rh; 8100 cfm of outdoor air mixes with 18,900 cfm of recirculated air. The mixture is heated by a steam coil and humidified by a pan humidifier to final conditions of 115°F DB and 20% rh.

1. What steam flow, in pounds per hour, should be supplied to the heating coil?
2. Estimate the steam consumption of the humidifier.

Problem 6 Outdoor air (8000 cfm) at 10°F DB and 50% rh enters the central apparatus of a split heating system. It is tempered to 55°F DB. Then, it flows through a spray humidifier where the leaving sump water is maintained at 50°F. The spray humidifier has a performance factor of 0.80. After leaving the humidifier, the air flows through a steam heating coil and is heated to 70°F DB.

1. What is the final relative humidity and humidity ratio of the air as it leaves the heating coil?
2. Assume steam at 2 psig and 90% quality is supplied to the tempering coil, the sump water heat exchanger, and the heating coil. How many pounds of steam per hour should be supplied to each?

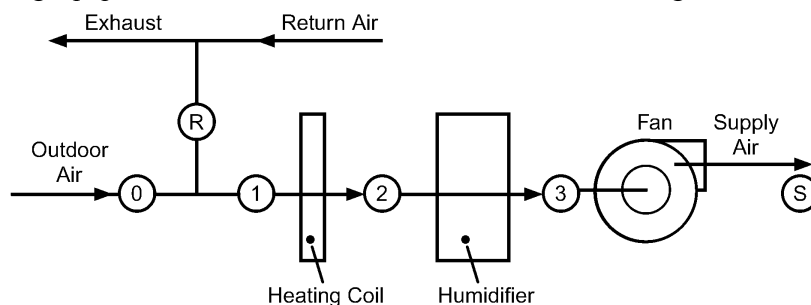
Problem 7 The heat exchanger for the spray water in Problem 6 is out of service for maintenance. The split heating system is operating as specified except that the sump water is recirculated. Assume makeup water to the sump is 37°F and saturating effectiveness is equal to the performance factor.

1. What is the final relative humidity and humidity ratio of the air leaving the heating coil?
2. What is the steam rate (lb/h) for the tempering coils and for the heating coil?

Problem 8 Air at 105°F DB (40.6°C) and 75% rh passes through a chilled water spray. Air leaves the spray chamber at 45°F DB (7.2°C) saturated. How many grains of moisture per pound of entering air are condensed?

Problem 9 A building space is to be maintained at 70°F and 35% rh when outdoor design temperature is 10°F. Design heat losses from the space are 250,000 Btu/h, sensible, and 45,000 Btu/h, latent. Ventilation requires that 1500 cfm of outdoor air be used. Supply air is to be at 120°F.

The conditioning equipment and nomenclature are shown in the following sketch.



Determine the following:

1. The amount of supply air required, lb/h and cfm
2. The capacity of the heating coil, Btu/h, if
 - a. The humidifier is a spray washer using recirculated spray water with makeup water provided at 60°F
 - b. The humidifier is a steam humidifier using dry, saturated steam at 17.2 psia
3. The capacity of the humidifier, lb/h.

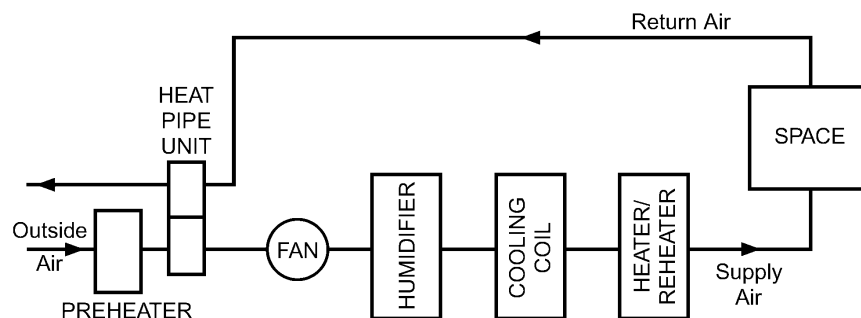
Problem 10 A spray-type air washer is to be used for humidification as well as cleaning of 9000 scfm of air. Inlet conditions to the washer are 75°F DB and 48 F WB. Desired humidity ratio at outlet is 0.005 lb_w/lb_a. Determine the following:

1. The necessary humidification efficiency of the washer, %
2. The makeup water requirements (humidifying capacity) of the unit, lb_w/h

Problem 11 A heat pipe air-to-air energy recovery device is being considered for a system requiring 9000 scfm of outdoor air. Initially, a separate preheater was planned for bringing the outdoor air from its −2°F design ambient outdoor temperature to 40°F. Determine the following:

1. The rating (Btu/h)
2. The sensible effectiveness (%) to specify for the heat pipe unit if it is to eliminate the need for the air preheater

Problem 12 The HVAC system for a hospital operating room, which requires 100% outdoor air, is shown in the following figure and includes an air-to-air heat pipe energy recovery unit having a sensible effectiveness of 73%. The air leaving the cooling coil is maintained at 58°F, 90% rh, all year long. During winter operation, air leaves the heater at 130°F. Fan speed is changed between summer and winter operation. Design duct system pressure drop (summer) is 3.25 in. water.



1. At winter design conditions (indoor: 72°F and 30% rh; outdoor: 5°F and 100% rh) the space load is 235,000 Btu/h (sensible) with negligible latent load. Determine (a) the necessary size of heating unit (Btu/h) both with and without the energy recovery unit and (b) the humidifier size (gal/day). Neglect fan effects.
2. At summer design conditions (indoor: 78°F; outdoor: 95°F DB/76°F WB), the space cooling loads are 146,000 Btu/h (sensible) and 79,000 Btu/h (latent). Determine (a) fan size (hp and scfm), (b) sensible coil load, Btu/h, (c) latent coil load, Btu/h, and (d) necessary size of cooling unit, Btu/h, both with and without the energy recovery unit. Include fan effects.

SI Resource

Table 2 Comparison of Air-to-Air Energy Recovery Devices (SI)
(Table 3, Chapter 26, 2016 *ASHRAE Handbook—HVAC Systems and Equipment*, SI Version)

	Fixed Plate	Membrane Plate	Energy Wheel	Heat Wheel	Heat Pipe	Runaround Coil Loop	Thermosiphon	Twin Towers
Airflow arrangements	Counterflow Cross-flow	Counterflow Cross-flow	Counterflow Parallel flow	Counterflow	Counterflow Parallel flow	—	Counterflow Parallel flow	—
Equipment size range, cfm (L/s)	50 and up (25 and up)	50 and up (25 and up)	50 to 74,000 and up (25 to 35 000 and up)	50 to 74,000 and up (25 to 35 000 and up)	100 and up (50 and up)	100 and up (50 and up)	100 and up (50 and up)	—
Typical sensible effectiveness ($m_s = m_e$), %	50 to 80	50 to 75	50 to 85	50 to 85	45 to 65	45 to 65	40 to 60	40 to 60
Typical latent effectiveness,* %	—	50 to 72	50 to 85	0	—	—	—	—
Total effectiveness,* %	—	50 to 73	50 to 85	—	—	—	—	—
Face velocity, fpm (m/s)	200 to 1000 (1 to 5)	200 to 600 (1 to 3)	500 to 1000 (2.5 to 5)	400 to 1000 (2 to 5)	400 to 800 (2 to 4)	300 to 600 (1.5 to 3)	400 to 800 (2 to 4)	300 to 450 (1.5 to 2.2)
Pressure drop, in. of water (Pa)	0.4 to 4 (100 to 1000)	0.4 to 2 (100 to 500)	0.4 to 1.2 (100 to 300)	0.4 to 1.2 (100 to 300)	0.6 to 2 (150 to 500)	0.6 to 2 (150 to 500)	0.6 to 2 (150 to 500)	0.7 to 1.2 (170 to 300)
EATR, %	0 to 2	0 to 5	0.5 to 10	0.5 to 10	0 to 1	0	0	0
OACF	0.97 to 1.06	0.97 to 1.06	0.99 to 1.1	1 to 1.2	0.99 to 1.01	1.0	1.0	1.0
Temperature range, °F (°C)	−75 to 1470 (−60 to 800)	−40 to 120 (−40 to 50)	−65 to 1470 (−55 to 800)	−65 to 1470 (−55 to 800)	−40 to 104 (−40 to 40)	−50 to 930 (−45 to 500)	−40 to 104 (−40 to 40)	−40 to 115 (−40 to 46)
Typical mode of purchase	Exchanger only Exchanger in case Exchanger and blowers Complete system	Exchanger only Exchanger in case Exchanger and blowers Complete system	Exchanger only Exchanger in case Exchanger and blowers Complete system	Exchanger only Exchanger in case Exchanger and blowers Complete system	Exchanger only Exchanger in case Exchanger and blowers Complete system	Coil only Complete system	Exchanger only Exchanger in case	Complete system
Advantages	No moving parts Low pressure drop Easily cleaned	No moving parts Low pressure drop Low air leakage Moisture/mass transfer	Moisture/mass transfer Compact large sizes Low pressure drop Easily cleaned Available on all ventilation system platforms	Compact large sizes Low pressure drop Easily cleaned	No moving parts except tilt Fan location not critical Allowable pressure differential up to 2 psi (15 kPa)	Exhaust air-stream can be separated from supply air Fan location not critical	No moving parts Exhaust air-stream can be separated from supply air Fan location not critical	Latent transfer from remote airstreams Efficient micro-biological cleaning of both supply and exhaust airstreams
Limitations	Large size at higher flow rates	Few suppliers Long-term maintenance and performance unknown	Supply air may require some further cooling or heating Some EATR without purge	Some EATR with purge	Effectiveness limited by pressure drop and cost Few suppliers	Predicting performance requires accurate simulation model	Effectiveness may be limited by pressure drop and cost Few suppliers	Few suppliers Maintenance and performance unknown
Heat rate control (HRC) methods	Bypass dampers and ducting	Bypass dampers and ducting	Bypass dampers and wheel speed control	Bypass dampers and wheel speed control	Tilt angle down to 10% of maximum heat rate	Bypass valve or pump speed control	Control valve over full range	Control valve or pump speed control over full range

*Rated effectiveness values are for balanced flow conditions. Effectiveness values increase slightly if flow rates of either or both airstreams are higher than flow rates at which testing is done.

EATR = exhaust air transfer ratio
OACF = outdoor air correction factor