

CHAPTER 41. EVAPORATIVE AIR-COOLING EQUIPMENT

THIS chapter addresses direct and indirect evaporative equipment, air washers, and their associated equipment used for air cooling, humidification, dehumidification, and air cleaning. Humidification equipment for residential and industrial applications is covered in [Chapter 22](#). Principal advantages of evaporative air conditioning include

- Substantial energy and cost savings
- Reduced peak power demand
- Improved indoor air quality
- Life-cycle cost effectiveness
- Easily integrated into built-up systems
- Wide variety of packages available
- Provides humidification and dehumidification when needed
- Easy to use with direct digital control (DDC)
- Reduced pollution emissions
- No use of chlorofluorocarbons (CFCs)
- For same amount of cooling, less water is evaporated than with conventional air conditioning
- Sound attenuation

Packaged direct evaporative air coolers, air washers, indirect evaporative air coolers, evaporative condensers, vacuum cooling apparatus, and cooling towers exchange sensible heat for latent heat. This equipment falls into two general categories: those for (1) air cooling and (2) heat rejection. This chapter addresses equipment used for air cooling. Adiabatic evaporation of water provides the cooling effect of evaporative air conditioning. In **direct evaporative cooling**, water evaporates directly into the airstream, reducing the air's dry-bulb temperature and raising its humidity level. Direct evaporative equipment cools air by direct contact with the water, either by an extended wetted-surface material (e.g., packaged air coolers) or with a series of sprays (e.g., an air washer). With **indirect evaporative cooling**, secondary air removes heat from primary air using a heat exchanger. In one indirect method, water is evaporatively cooled by a cooling tower and circulates through one side of a heat exchanger. Supply air to the space passes over the other side of the heat exchanger. In another common method, one side of an air-to-air heat exchanger is wetted and removes heat from the conditioned supply airstream on the dry side. Even in regions with high wet-bulb temperatures, indirect evaporative cooling can be economically feasible. This is especially true if building return air from an air-conditioned building is used on the wet side of an air-to-air heat exchanger. The return air's lower wet-bulb temperature, which derives from mechanical refrigeration, may be used to extend indirect evaporative cooling performance in more humid climates. To improve system performance, it is often desirable to combine the effects of both indirect (dry) and direct (adiabatic) evaporative cooling. When indirect cooling is called for as the first stage of cooling, outdoor air is sensibly cooled and the ambient wet-bulb temperature is reduced. At this new, lower wet-bulb condition, the direct evaporative cooler (second stage) can generate a lower delivery dry-bulb temperature. If a final cooling stage of refrigeration is required to achieve a 55°F building variable-air-volume (VAV) delivery temperature, final-stage cooling capacity may be greatly reduced when compared to an air-side economizer with 25% minimum outdoor air. In semiarid climates, where most ambient dew-point conditions are below 55°F, two-stage indirect/direct evaporative cooling may reduce peak building demand by 25 to 30% and ton-hours of cooling by 50 to 60%. During cooling hours, the indirect/direct cooling system provides 100% outdoor air to the building for better indoor air quality (IAQ). Using building return air on the secondary (wet) side of a directly sprayed air-to-air heat exchanger has other benefits. With a 55°F VAV building delivery set-point temperature, return air wet-bulb temperatures are commonly between 60 and 64°F because of building moisture capacitance. In summer, using building return air wet bulb for indirect evaporative cooling ensures higher sensible cooling from the indirect cooling heat exchanger, especially in climates where ambient wet-bulb excursions above 65°F are frequent. In winter, when the air-to-air heat exchanger water sprays are off, a building return air dry-bulb condition of 70 to 75°F may be used for heat recovery from outdoor air. VAV systems, which experience reduced outdoor air as supply fans turn down, can provide a higher percentage of outdoor air to comply with ASHRAE *Standard* 62.1 without added heating costs. Direct evaporative coolers for residences in low-wet-bulb regions typically require 70% less energy than direct-expansion air conditioners. For instance, in El Paso, Texas, the typical evaporative cooler consumes 609 kWh per cooling season, compared to 3901 kWh per season for a typical vapor-compression air conditioner with a seasonal energy-efficiency ratio (SEER) of 10. This equates to an average demand of 0.51 kW based on 1200 operating hours, compared to an average of 3.25 kW for a vapor-compression air conditioner. Depending on climatic conditions, many buildings can use indirect/direct evaporative air conditioning to provide comfort cooling. Indirect/direct systems achieve a 40 to 50% energy savings in moderate humidity zones (Foster and Dijkstra 1996).

1. DIRECT EVAPORATIVE AIR COOLERS

With direct evaporative air cooling, air is passed through porous wetted pads or a spray, and its sensible heat energy evaporates some water. Heat and mass transfer between the air and water results in an adiabatic process that lowers the dry-bulb temperature while increasing the air's moisture content (the wet-bulb temperature remains nearly constant). The dry-bulb temperature of the nearly saturated air approaches the ambient air's wet-bulb temperature. Saturation effectiveness is a key factor in determining evaporative cooler performance. The extent to which the leaving air temperature from a direct evaporative cooler approaches the thermodynamic wet-bulb temperature of the entering air defines the **direct saturation efficiency** ϵ_s , expressed as

$$\epsilon_s = 100 \frac{t_1 - t_2}{t_1 - t_s'} \tag{1}$$

where

- ϵ_s = direct evaporative cooling saturation efficiency, %
 - t_1 = dry-bulb temperature of entering air, °F
 - t_2 = dry-bulb temperature of leaving air, °F
 - t_s' = thermodynamic wet-bulb temperature of entering air, °F
- An efficient wetted pad (with high saturation efficiency) can reduce the air dry-bulb temperature by as much as 95% of the wet-bulb depression (ambient dry-bulb temperature less wet-bulb temperature), although an inefficient and poorly designed pad may only reduce this by 50% or less. Although direct evaporative cooling is simple and inexpensive, its cooling effect is not applicable for indoor comfort when the ambient wet-bulb temperature is higher than about 70°F; however, this cooling is satisfactory for relief cooling applications (e.g., greenhouses, industrial cooling). Direct evaporative coolers should not recirculate indoor air; exhaust should equal incoming conditioned air.

Random-Media Air Coolers

These coolers contain evaporative pads, usually of aspen wood or absorbent plastic foam ([Figure 1](#)). A water-recirculating pump lifts sump water to a distributing system, and it flows down through the pads back to the sump. A fan in the cooler draws air through the evaporative pads and delivers it for space cooling. The fan discharges either through the side of the cooler cabinet or through the sump bottom. Random-media packaged air coolers are made as small tabletop coolers (50 to 200 cfm), window units (100 to 4500 cfm), and standard duct-connected coolers (5000 to 18,000 cfm). Cooler selection is based on a capacity rating from an independent agency.

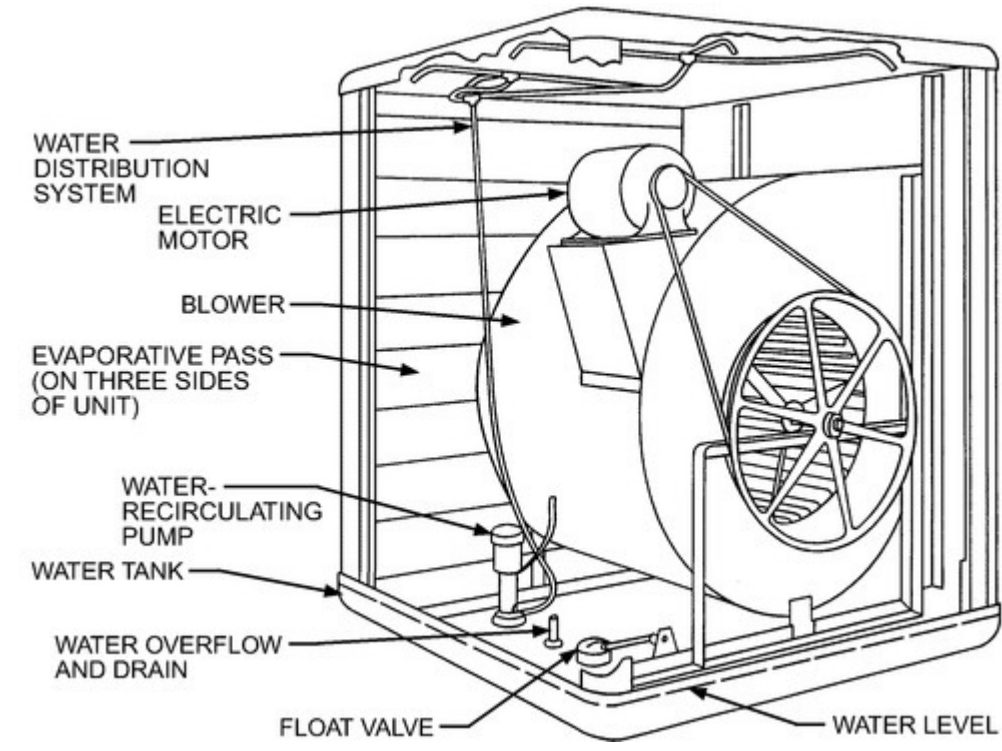


Figure 1. Typical Random-Media Evaporative Cooler

When clean and well maintained, commercial random-media air coolers operate at approximately 80% effectiveness and reduce the concentration of 10 µm and larger particles in the air. In some units, supplementary filters are added to reduce the particle count of delivered air when the unit is operating with or without water circulation. Evaporative pads may be chemically treated to increase wettability. An additive may be included in the fibers to help them resist attack by bacteria, fungi, and other microorganisms. Random-media cooler designs with face velocity of 100 to 250 fpm with a pressure drop of 0.1 in. of water are the norm. Aspen fibers packed to approximately 0.3 to 0.4 lb/ft² of face area, based on a 2 in. thick pad, are standard. Pads mount in removable louvered frames, which are usually made of painted galvanized steel or molded plastic. Troughs distribute water to the pads. A centrifugal pump with a submerged inlet delivers water through tubes that provide an equal flow of water to each trough. It is important that the pump motor be thermally protected. The sump or water tank has a water makeup connection, float valve, overflow pipe, and drain. It is important to provide bleed water or a timed dump of the sump (or both) to prevent build-up of minerals, dirt, and microbial growth. The fan is usually a forward-curved, centrifugal fan, complete with motor and drive. The V-belt drive may include an adjustable-pitch motor sheave to allow fan speed to increase to use the full motor capacity at higher airflow resistance. The motor enclosure may be drip-proof, totally enclosed, or a semi-open type specifically designed for evaporative coolers.

Rigid-Media Air Coolers

Blocks of corrugated material make up the wetted surface of rigid-media direct evaporative air coolers ([Figure 2](#)). Materials include cellulose, plastic, and fiberglass, treated to absorb water and resist weathering effects. The medium is cross corrugated to maximize mixing of air and water. In the direction of airflow, the depth of medium is commonly 12 in., but it may be between 4 and 24 in., depending on the desired thermal performance. The medium has the desirable characteristics of low resistance to airflow, high saturation effectiveness, and self-cleaning abilities. The standard design face velocity of a rigid medium is 400 to 600 fpm. Static pressure loss for a 12 in. media pad varies from 0.14 to 0.3 in. of water at sea level. Direct evaporative air coolers using this material can handle as much as 600,000 cfm and may include an integral fan. Saturation effectiveness varies from 70 to over 95%, depending on media depth and air velocity. Air flows horizontally while the recirculating water flows vertically over the medium surfaces by gravity feed from a flooding header and water distribution chamber. The header may be connected directly to a pressurized water supply for once-through operation (e.g., gas turbines, cleanrooms, data centers), or a pump may recirculate the water from a lower reservoir constructed of heavy-gauge corrosion-resistant material. The reservoir is also fitted with overflow and positive-flowing drain connections. The upper media enclosure is of reinforced galvanized steel or other corrosion-resistant sheet metal, or of plastic.

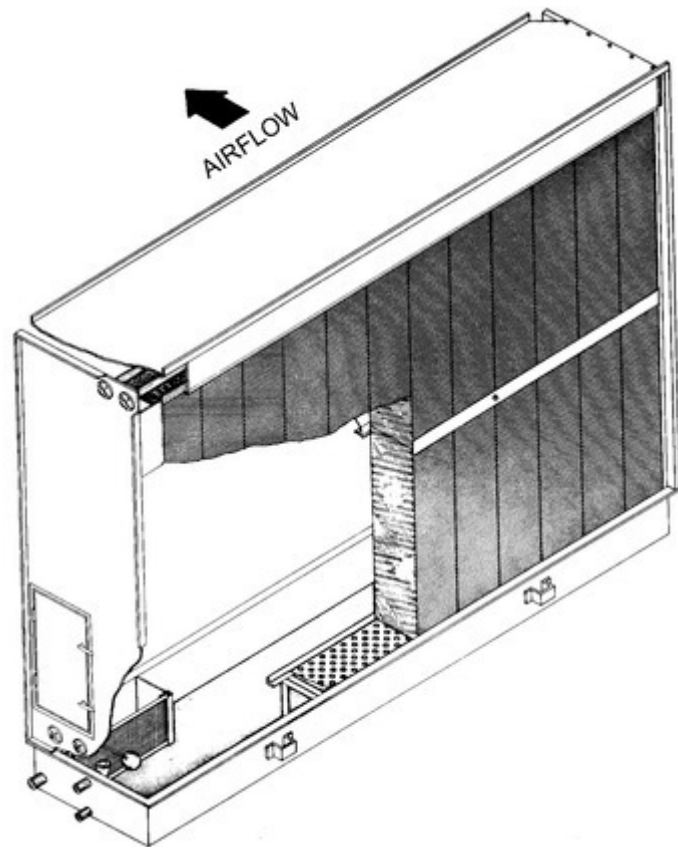


Figure 2. Typical Rigid-Media Air Cooler

Flanges at the entering and leaving faces allow connection of ductwork. In recirculating water systems, a float valve maintains proper water level in the reservoir, makes up water that has evaporated, and supplies fresh water for dilution to prevent an overconcentration of solids and minerals. Because the water recirculation rate is low and high-pressure nozzles are not needed to saturate the medium, pumping power is low compared to spray-filled air washers with equivalent evaporative cooling effectiveness.

Remote Pad Evaporative Cooling Equipment

Greenhouses, poultry or hog buildings, and similar applications use exhaust fans installed in the wall or roof of the structure. Air evaporatively cools as it flows through pads located on the other end of the building. Water flowing down from a perforated pipe wets the pads, with excess water collected for recirculation. In some cases, the pads are wetted with high-pressure fogging nozzles, which provide additional cooling. Water for fogging nozzles must come directly from the fresh-water supply. The pad has an air velocity of approximately 150 fpm for random-media pads, 250 fpm for 4 in. rigid media, and 425 fpm for 6 in. rigid media.

2. INDIRECT EVAPORATIVE AIR COOLERS

Packaged Indirect Evaporative Air Coolers

[Figure 3](#) shows an indirect evaporative cooling (IEC) heat exchanger using elliptical shaped polymer tubes installed horizontally. Either outdoor or recirculated air may be cooled using IEC; commercial IEC commonly cools outdoor air, often as part of a dedicated outdoor-air system (DOAS) or VAV air-handler design, whereas in industrial applications and data center cooling designs, process recirculation air is more commonly cooled. Air delivered to the room or process flows through the inside of the tubes, where it is sensibly cooled. A sump pump recirculates water over the outside of the polymer tubes of this cross-flow air-to-air heat exchanger. Scavenger outdoor air or building return/exhaust air (typical for commercial applications) is drawn upward over the exterior of the wetted polymer tubes.

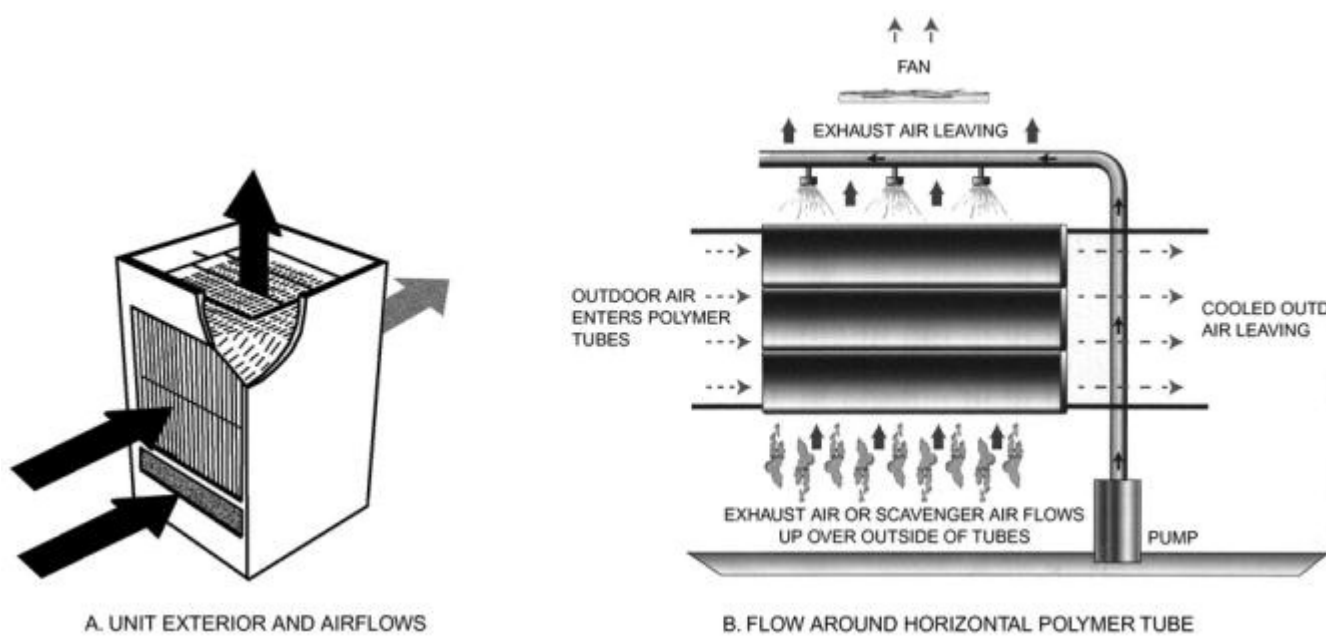


Figure 3. Indirect Evaporative Cooling (IEC) Heat Exchanger (Courtesy Munters/Des Champs)

The airfoil elliptical shape of the tube exterior maximizes the heat transfer surface area. Sensible heat from process air flowing inside the tubes is transferred through the thin polymer tube wall into the water film flowing over the exterior of the tubes. The water is cooled by evaporation as the scavenger air flows over the wetted tube exterior. The scavenger air, along with the extracted heat, is finally discharged to ambient. Process air inside the tubes may be dehumidified whenever the wet-bulb temperature of the scavenger air is below the dew-point temperature of the process air. As water evaporates on the exterior of the tubes, it is normal for some minerals in the water to be deposited onto the tube walls. Fortunately, the ribbed exterior of properly design polymer tubes resists mineral adhesion. Also, elliptical polymer tubes are elastic enough that normal fan operation for supply and scavenger air flexes the tubes, helping them shed scale accumulations and minimizing fouling so as not to adversely affect IEC performance. Polymer tube heat exchangers can provide a 60 to 80% approach of the dry-side entering dry bulb temperature to the wet-side entering wet bulb temperature. The heat exchanger efficiency calculation is called **wet-bulb depression efficiency (WBDE)** and is defined as

$$WBDE = 100 \frac{(t_1 - t_2)}{(t_1 - t_s')} \tag{2}$$

where

- WBDE = wet-bulb depression efficiency, %
 - t_1 = dry-bulb temperature of entering primary air, °F
 - t_2 = dry-bulb temperature of leaving primary air, °F
 - t_s' = wet-bulb temperature of entering secondary air, °F
- Supply-air-side static pressure losses for these heat exchangers range from 0.25 to 0.75 in. of water. Wet-side airflow pressure drop penalties range from 0.4 to 0.9 in. of water. Secondary airflow ratios are in the range of 1.5 to 1 down to a low of 1 cfm of outdoor air (OA) to 0.7 cfm of secondary airflow. The higher the ratios of wet-side air to dry air, the greater the WBDE, with all other factors remaining constant. Cooling energy efficiency ratios (EER) for this type of heat exchanger range from 40 to 80. With **DX refrigeration**, [Figure 4](#) shows a package unit design that combines the tube-type indirect evaporative cooling heat exchanger with a direct-expansion (DX) refrigeration final stage of cooling. The geometry of the tube-type heat exchanger usually limits the size of this application to less than 50,000 cfm of supply air for complete packaged systems. There is no airflow limit for stand-alone tube-type IEC heat exchangers that are banked together (e.g., in a mechanical room on the outer wall of a large data center). By placing the condenser coil in the wet-side air path leaving the heat exchanger, the mechanical cooling component's coefficient of performance (COP) significantly increases over that of an air-cooled condenser system with the coil in the ambient air. When building return air is used as the secondary airflow, compressor energy inputs are often reduced from 1.1 kW per ton to 0.70 kW per ton or lower, because building return air from an air-conditioned building has wet-bulb conditions in the range of 60 to 65°F at a 75°F room dry-bulb temperature. Wet-side air leaving the heat exchanger is usually in the range of 70 to 75°F db, but at 80 to 90% rh, depending on the heat exchanger's wetting efficiency. Because refrigeration air-cooled condenser coils are unaffected by humidity, this cooler airstream may be used to reduce the refrigeration condensing temperature of the DX system, which increases compressor capacity and life by reducing vapor compression temperature lift.

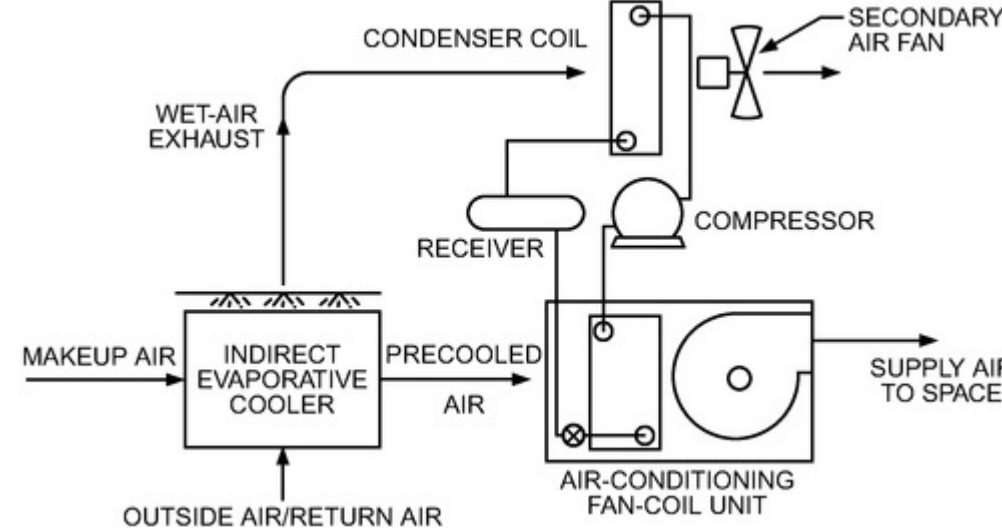


Figure 4. Indirect Evaporative Cooler Used as Precooler

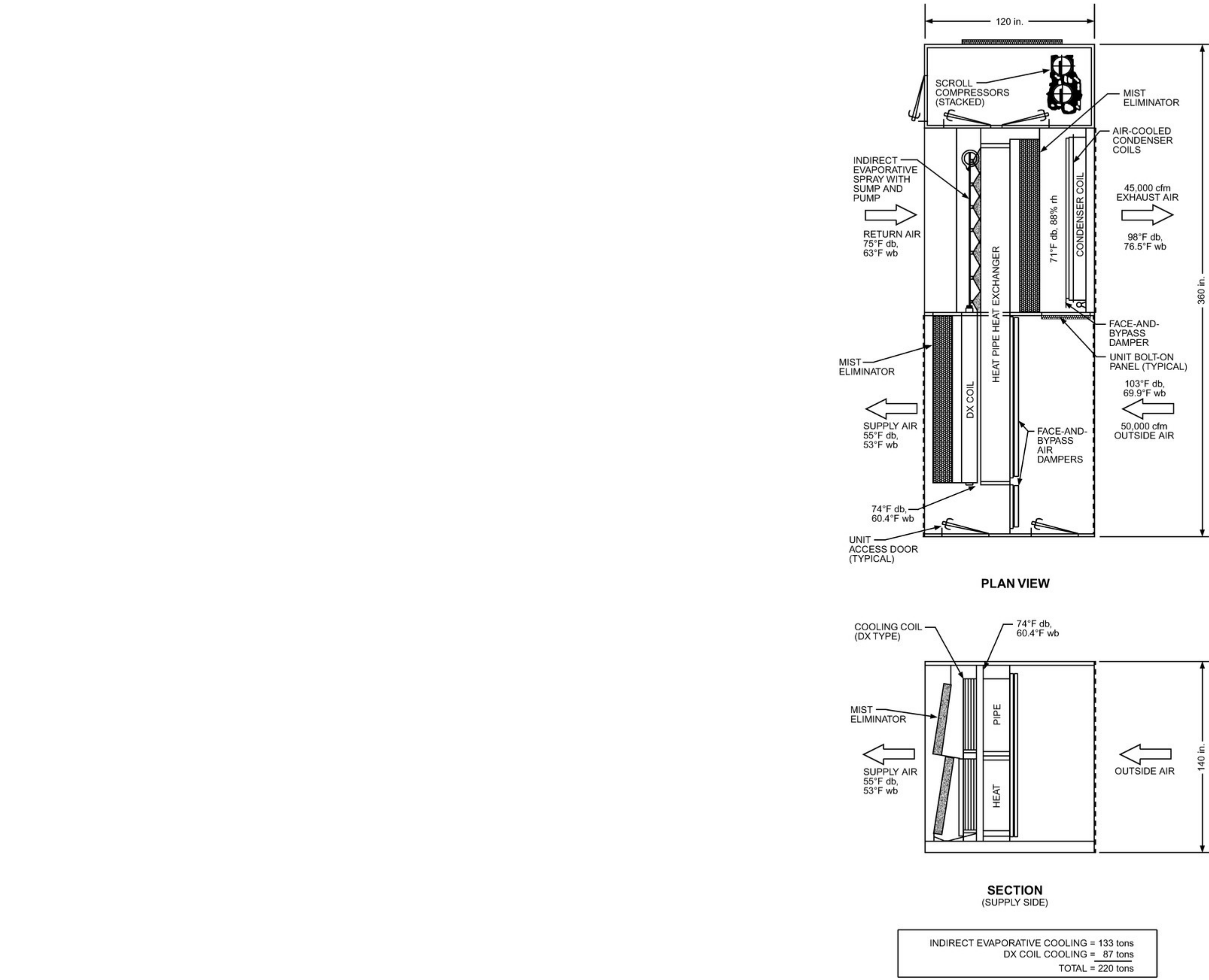


Figure 5. Heat Pipe Indirect Evaporative Cooling (IEC) Heat Exchanger Packaged with DX System

Figure 5 shows how a heat-pipe, indirect evaporative cooling heat exchanger may be packaged with a DX-type refrigeration system, using building return air, to minimize cooling energy consumption for an all-outdoor-air design such as may be required for a laboratory or hospital application. The geometry of the heat pipe lends itself to the treatment of larger airflow quantities. The dimensions shown in Figure 5 are for a nominal 50,000 cfm supply air system with 220 tons of total load. In addition, the heat pipe heat exchanger has the distinct advantage over other air-to-air heat exchangers of being able to isolate contaminated exhaust air from clean makeup air with a double-walled partition at the center bulkhead separating the two airflow. For laboratory applications, supply air fans should be positioned to blow through the heat pipe, to allow the heat pipe indirect evaporative cooler to remove some of the supply fan heat from the air before its delivery to the DX evaporator coil.

As an example, Figure 5 shows state-point conditions at each stage of the process, assuming a required 55°F db supply air temperature and an outdoor air (OA) inlet condition at summer design of 103°F db and 69.9°F wb. The indirect-cooling heat pipe reduces the outdoor air to 74°F db, 60.4°F wb where it enters the direct-expansion (DX) cooling evaporator coil. The refrigeration coil sensibly cools the outdoor air to 55°F db and 53°F wb, which for 50,000 cfm would require 1,045,000 Btu/h or 87 tons. All values are for a sea-level application.

On the return-air side of the heat pipe heat exchanger, the condition entering the heat pipe is 75°F db and 63°F wb. After passing through the wet side of the heat pipe, the return air enters the condenser coil at 71°F and 88% rh. The heat of compression (110 tons) is rejected to the 45,000 cfm airflow and exhausted at a condition of 98°F db, 76.5°F wb.

A mist eliminator downstream of the sprayed heat pipe keeps water droplets from carrying over to wet the refrigerant-condensing coil. This cool, humid exhaust air provides an excellent source into which the condenser coil may reject heat. Condenser coil face-and-bypass dampers control condensing head pressure within an acceptable range. During winter, when the heat exchanger recovers heat and the sprays are off, these dampers are both open to minimize the condenser coil static pressure penalty.

Many applications below 200 tons use roof-mounted, air-cooled condensers. The 50,000 cfm IEC unit in Figure 5 delivers 133 tons of sensible cooling to the outdoor air with an energy consumption of 0.2 kW per ton and an EER of 60. The evaporatively cooled refrigeration provides the remaining 87 tons of cooling required on the hottest day of the summer. To deliver 55°F db and 53°F wb to the building, the energy consumed for the refrigeration component is 0.7 kW per ton, with an EER of 17.1. A conventional air-cooled condensing unit on the roof in 100°F ambient temperatures typically requires 1.1 kW per ton to deliver 220 tons of total load, or a total peak demand of 242 kW. By comparison, on the hottest day of the year, the heat-pipe IEC and evaporatively cooled refrigeration design only consume 87.55 kW for a combined EER of 30.2. The total peak demand reduction for an all-outdoor-air design in this example is 154.45 kW.

Because the wet side of the heat pipe has a surface temperature of 70 to 75°F when subjected to 100°F ambient air temperatures, scale and fouling of the exhaust-side surface progress very slowly. Systems of this type have been in successful service for over 25 years at various sites in North America.

For sprayed heat-pipe applications, a one-piece heat pipe is recommended. All-aluminum heat pipes are available constructed of series 3003 alloy. The fin surface is extruded directly from the heat tube wall. Corrosion-resistant coating for the wet-side surface may be necessary in some hard-water applications. Wastewater bleed rates should be field set based on the water chemistry analysis. Water consumption in the range of 1 to 1.5 gpm per 10,000 cfm of supply air is typical, for both evaporation and bleed, for an IEC system.

Chapter 52 of the 2015 ASHRAE Handbook—HVAC Applications includes sample evaporative cooling calculations. Manufacturers' data should be followed to select equipment for cooling performance, pressure drop, and space requirements.

Manufacturers' ratings require careful interpretation. The basis of ratings should be specified because, for the same equipment, performance is affected by changes in primary and secondary air velocities and mass flow ratios, wet-bulb temperature, altitude, and other factors.

Typically, air resistance on both primary and secondary sections ranges between 0.2 and 2.0 in. of water. The ratio of secondary air to conditioned primary air may range from less than 0.3 to greater than 1.0, and has an effect on performance (Peterson 1993). Based on manufacturers' ratings, available equipment may be selected for indirect evaporative cooling effectiveness ranging from 40 to 80%.

Heat Recovery

Indirect evaporative cooling has been used in a number of heat recovery systems, including plate heat exchangers (Scoffield and DesChamps 1984), heat pipe heat exchangers (Mathur 1991; Scofield 1986), rotary regenerative heat exchangers (Woolridge et al. 1976), and two-phase thermosiphon loop heat exchangers (Mathur 1990). Indirect evaporative cooling/heat recovery can be retrofitted on existing systems, lowering operational cost and peak demand (Goswami and Mathur 1993, 1995). For new installations, equipment can be downsized, lowering overall project and operational costs. Chapter 26 has more information on using indirect evaporative cooling with heat recovery.

Cooling Tower/Coil Systems

Combining a cooling tower or other evaporative water cooler with a water-to-air heat exchanger coil and water-circulating pump is another type of indirect evaporative cooling. Water flows from the cooling tower reservoir to the coil and returns to the tower's upper distribution header. Both open-water and closed-loop systems are used. Coils in open systems should be cleanable. Recirculated water evaporatively cools to within a few degrees of the wet-bulb temperature as it flows over the wetted surfaces of the cooling tower. As cooled water flows through the tubes of the coil in the conditioned airstream, it picks up heat from the conditioned air. The water temperature increases, and the primary air is cooled without adding moisture to it. The water again cools as it recirculates through the cooling tower. A float valve controls the fresh-water makeup, which replaces evaporated water. Bleedoff prevents excessive concentration of minerals in recirculated water.

One advantage of a cooling tower, especially for retrofits, large built-up systems, and dispersed air handlers, is that it may be remotely located from the cooling coil. In addition, the tower is more accessible for maintenance. Overall WBDE may range between 55% and 75% or higher. If return air goes to the cooling tower of an indirect cooling system before discharging outdoors, the cooling tower should be specifically designed for this purpose. These coolers wet a medium that has a high ratio of wetted surface area per unit of medium volume. Performance depends on depth of the medium, air velocity over the medium surface, water flow to airflow ratio, wet-bulb temperature, and water-cooling range. Because of the close approach of the water temperature to the wet-bulb temperature, overall effectiveness may be higher than that of a conventional cooling tower.

Other Indirect Evaporative Cooling Equipment

Other combinations of evaporative coolers and heat exchangers can accomplish indirect evaporative cooling. Heat pipes and rotary heat wheels, two-phase thermosiphon coil loops, plate and pleated media, and shell-and-tube heat exchangers have all been used. If the conditioned (primary) air and the exhaust or outdoor (secondary) airstream are side by side, a heat pipe or heat wheel can transfer heat from the warmer air to the cooler air. Evaporative cooling of the secondary airstream by spraying water directly on the surfaces of the heat exchanger (excluding heat wheels) or by a direct evaporative cooler upstream of the heat exchanger may cool the primary air indirectly by transferring heat from it to the secondary air.

3. INDIRECT/DIRECT COMBINATIONS

In a two-stage indirect/direct evaporative cooler, a first-stage indirect evaporative cooler lowers both the dry- and wet-bulb temperature of the incoming air. After leaving the indirect stage, the supply air passes through a second-stage direct evaporative cooler; Figure 6 shows the process on a psychrometric chart. First-stage cooling follows a line of constant humidity ratio because no moisture is added to the primary airstream. The second stage follows the wet-bulb line at the condition of the air leaving the first stage.

The indirect evaporative cooler may be any of the types described previously. Figure 7 shows a cooler using a rotary heat wheel or heat pipe. The secondary air may be exhaust air from the conditioned space or outdoor air. When secondary air passes through the direct evaporative cooler, evaporative cooling lowers the dry-bulb temperature. As this air passes through the heat wheel, the mass of the medium cools to a temperature approaching the wet-bulb temperature of the secondary air. The heat wheel rotates (note, however, that a heat pipe has no moving parts) so that its cooled mass enters the primary air and, in turn, sensibly cools the primary (supply) air. After the heat wheel or pipe, a direct evaporative cooler further reduces the dry-bulb temperature of the primary air. This method can lower the supply air dry-bulb temperature by 10°F or more below the initial secondary air wet-bulb temperature.

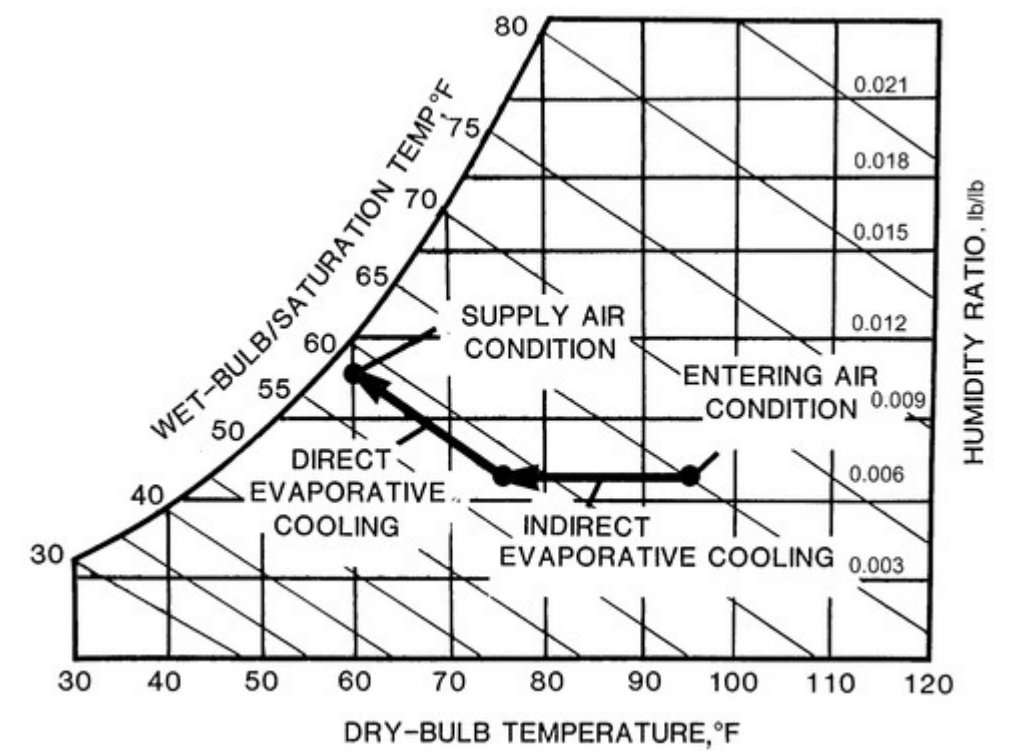


Figure 6. Combination Indirect/Direct Evaporative Cooling Process

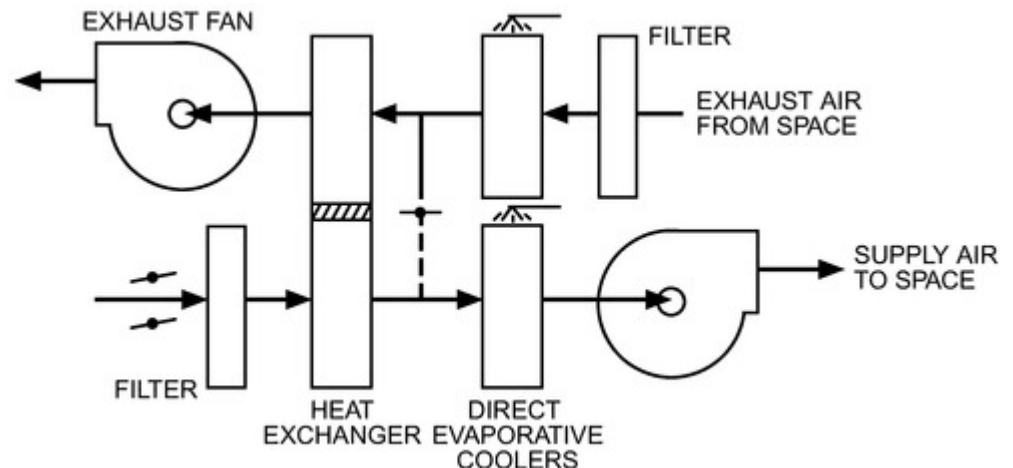


Figure 7. Indirect/Direct Evaporative Cooler with Heat Exchanger (Rotary Heat Wheel or Heat Pipe)

In areas where the 0.4% mean coincident wet-bulb design temperature is 66°F or lower, average annual cooling power consumption of indirect/direct systems may be as low as 0.22 kW/ton. When the 0.4% mean coincident wet-bulb temperature is as high as 74°F, indirect/direct cooling can have an average annual cooling power consumption as low as 0.81 kW/ton. By comparison, the typical refrigeration system with an air-cooled condenser may have average annual power consumption greater than 1.0 kW/ton.

In dry environments, indirect/direct evaporative cooling usually supplies 100% outdoor air to the conditioned spaces of a building. In these once-through applications, space latent loads and return air sensible loads are exhausted from the building rather than returned to the conditioning equipment. Consequently, the cooling capacity required from these systems may be less than from a conventional refrigerated cooling system. Design features to consider in systems such as the one in Figure 2 include air filters on the entering side of each heat wheel or pipe.

In areas with a high wet-bulb design temperature or where the design requires a supply air temperature lower than that attainable using indirect/direct evaporative cooling, a third cooling stage may be required. This stage may be a direct-expansion refrigeration unit or a chilled-water coil located either upstream or downstream from the direct evaporative cooling stage, but always downstream from the indirect evaporative stage. Refrigerated cooling occurs only when evaporative stages cannot achieve the required supply air temperature. Figure 8 shows a three-stage configuration (indirect/direct, with optional third-stage refrigerated cooling). The third-stage refrigerated cooling coil is downstream from the direct evaporative cooler. This requires careful selection and adjustment of controls to avoid removing more moisture by the refrigerated cooling coil than can be added by the direct evaporative cooling components. Analysis of static pressure drop through all components during design is critical to maintain optimum system total pressure loss and overall system efficiency. Note the face-and-bypass damper in Figure 8 around the indirect evaporative cooler. The bypass damper allows uncontaminated building return air to be recirculated in winter and mixed with outdoor air, as occurs in an air economizer. The fan parasitic losses of the indirect cooler heat exchanger may thus be reduced during cold weather for variable-air-volume air handlers.

The designer should consider using building exhaust and/or outdoor air as secondary air (whichever has the lower wet-bulb temperature) for indirect evaporative cooling. If possible, the indirect evaporative cooler should be designed to use both outdoor air and building exhaust as the secondary airstream; whichever source has the lower wet-bulb temperature should be used. Dampers and an enthalpy sensor are used to control this process. If the latent load in the space is significant, the wet-bulb temperature of the building exhaust air in cooling mode may be higher than that of the outdoor air. In this case, outdoor air may be used more effectively as secondary air to the indirect evaporative cooling stage.

Custom indirect/direct and three-stage configurations are available to allow many choices for location of the return, exhaust, and outdoor air; mixing of airstreams; bypass of components; or variable-volume control. Controllable elements include

- Modulating outdoor air and return air mixing dampers
- Secondary air fans and recirculating pumps of an indirect evaporative stage
- Recirculating pumps of a direct evaporative cooling stage
- Face-and-bypass dampers for the direct or indirect evaporative stage
- Chilled-water or refrigerant flow for a refrigerated stage
- System or individual terminal volume with variable-volume terminals, adjustable pitch fans, or variable-speed fans

For sequential control in indirect/direct evaporative cooling, the indirect evaporative cooler is energized for first-stage cooling, the direct evaporative cooler for second-stage cooling, and the refrigeration coil for third-stage cooling. In some applications, reversing the sequence of the direct and indirect evaporative coolers may reduce the first-stage power requirement. These systems are typically unfamiliar to most operations and maintenance staff, so special training may be needed.

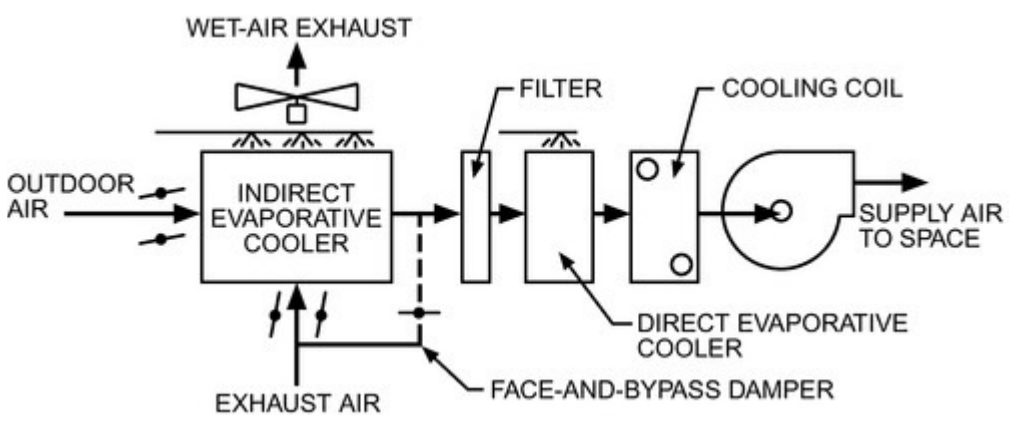


Figure 8. Three-Stage Indirect/Direct Evaporative Cooler

Precooling and Makeup Air Pretreatment

Evaporative cooling may be used to increase capacity and reduce the electrical demand of a mechanical refrigeration system. Both the condenser and makeup air may be evaporatively cooled by direct and/or indirect means.

The condenser may have its cooling enhanced by adding a direct evaporative cooler in front of the condenser coil. The direct evaporative cooler must add very little resistance to the airflow to the condenser, and face velocities must be well below velocities that would entrain liquid and carry it to the condenser. Condenser cooler maintenance should be infrequent and easy to perform. A well-designed direct evaporative cooler can reduce electrical demand and energy consumption of refrigeration units by 10 to 30%.

Makeup air cooling with an indirect/direct evaporative unit can be applied both to standard packaged units and to large built-up systems. Either outdoor air or building exhaust air (whichever has the lower wet-bulb temperature) can be used as the secondary air source. Outdoor air is generally easier to cool, and in some cases is the only option because the building exhaust is hazardous (e.g., from a laboratory) or remote from the makeup air inlet. If building exhaust air can be used as the secondary air source, it has the potential of heat recovery during cold weather. In general, outdoor air cooling has higher energy savings and lower electrical demand savings than return air cooling. These systems can significantly reduce the outdoor air load and should be analyzed using a psychrometric process for the region and climate being considered.

4. AIR WASHERS

Spray Air Washers

Spray air washers consist of a chamber or casing containing spray nozzles, a tank for collecting spray water as it falls, and an eliminator section for removing entrained drops of water from the air. A pump recirculates water at a rate higher than the evaporation rate. Intimate contact between the spray water and the air causes heat and mass transfer between the air and water (Figure 9). Air washers are commonly available from 2000 to 250,000 cfm capacity, but specially constructed washers can be made in any size. No standards exist; each manufacturer publishes tables giving physical data and ratings for specific products. Therefore, air velocity, water spray density, spray pressure, and other design factors must be considered for each application.

The simplest design has a single bank of spray nozzles with a casing that is usually 4 to 7 ft long. This type of washer is applied primarily as an evaporative cooler or humidifier. It is sometimes used as an air cleaner when the dust is wettable, although its air-cleaning efficiency is relatively low. Two or more spray banks are generally used when a very high degree of saturation is necessary and for cooling and dehumidification applications that require chilled water. Two-stage washers are used for dehumidification when the quantity of chilled water is limited or when the water temperature is above that required for the single-stage design. Arranging the two stages for water counterflow allows use of a small quantity of water with a greater water temperature rise.

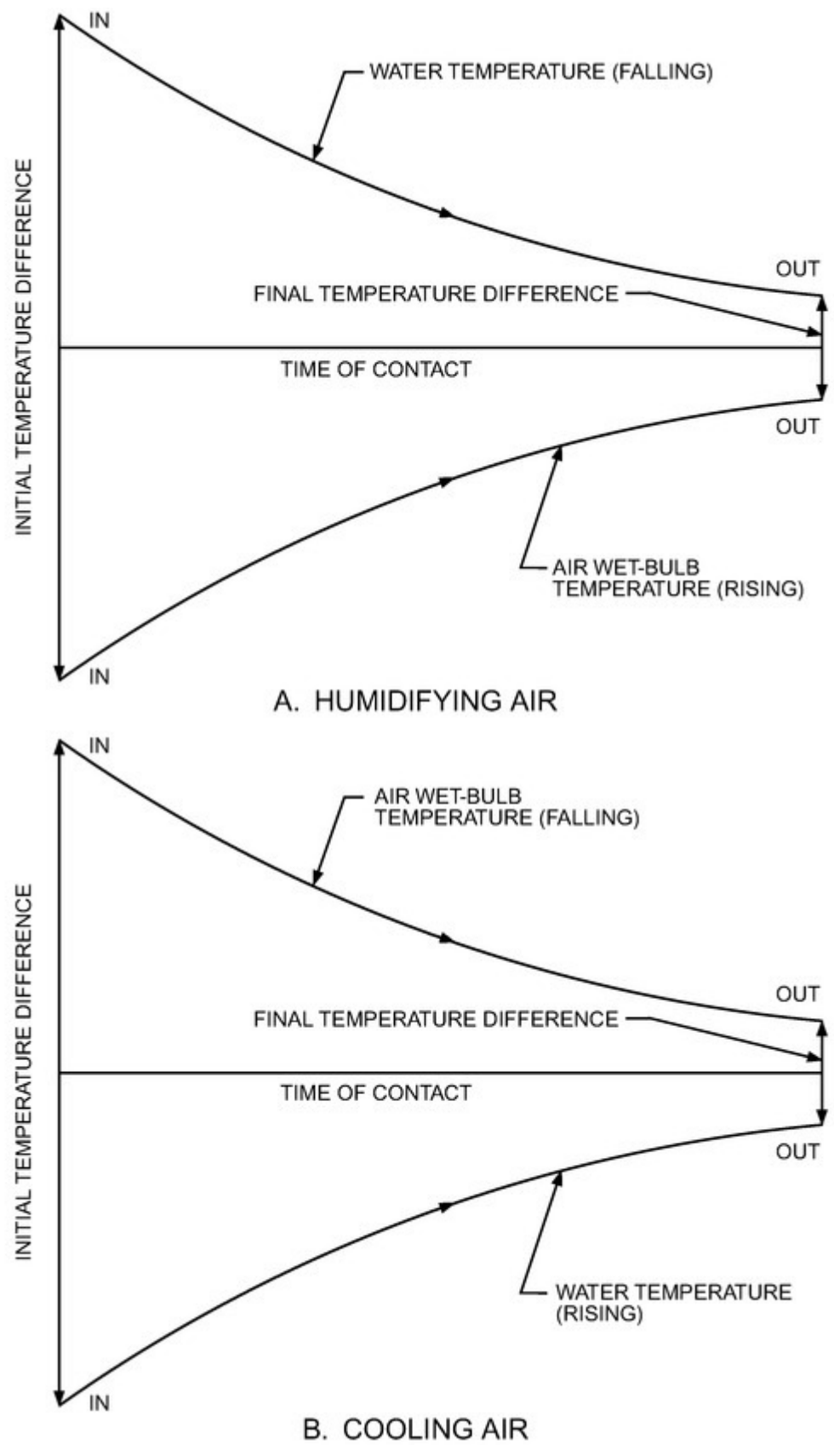


Figure 9. Interaction of Air and Water in Air Washer Heat Exchanger

Lengths of washers vary considerably. Spray banks are spaced from 2.5 to 4.5 ft apart; the first and last banks of sprays are located about 1 to 1.5 ft from the entering or leaving end of the washer. In addition, air washers may be furnished with heating or cooling coils in the washer chamber, which may affect the overall length of the washer. Some water (even very soft water) should always be bled off (continually and/or by using a dump or purge cycle) to prevent mineral build-up and to retard microbial growth. When the unit is shut down, all water should drain from the pipes. Low spots and dead ends must be avoided. Because an air washer is a direct-contact heat exchanger, water treatment is critical for proper operation as well as good hygiene. Algae and bacteria can be controlled by a chemical or ozone treatment program and/or regularly scheduled mechanical cleaning. Make sure that any chemicals used are compatible with all components in the air washer. Resistance to airflow through an air washer varies with the type and number of baffles, eliminators, and wetted surfaces; the number of spray banks and their direction and air velocity; the size and type of other components, such as cooling and heating coils; and other factors, such as air density. Pressure drop may be as low as 0.25 in. of water or as high as 1 in. of water. The manufacturer should be consulted regarding the resistance of any particular washer design combination. The casing and tank may be constructed of various materials. One or more doors are commonly provided for inspection and access. An air lock must be provided if the unit is to be entered while it is running. The tank is normally at least 16 in. high with a 14 in. water level; it may extend beyond the casing on the inlet end to make the suction strainer more accessible. The tank may be partitioned by a weir (usually in the entering end) to allow recirculation of spray water for control purposes in dehumidification work. The excess then returns over the weir to the central water-chilling machine. Eliminators consist of a series of vertical plates that are spaced about 0.75 to 2 in. on centers at the exit of the washer. The plates are formed with numerous bends to deflect air and obtain impingement on the wetted surfaces. Hooks on the edge of the plates improve moisture elimination. Perforated plates may be installed on the inlet end of the washer to obtain more uniform air distribution through the spray chamber. Louvers, which prevent backlash of spray water, may also be installed for this purpose.

High-Velocity Spray-Type Air Washers

High-velocity air washers generally operate at air velocities in the range of 1200 to 1800 fpm. Some have been applied as high as 2400 fpm, but 1200 to 1600 fpm is the most accepted range. The reduced cross-sectional area of high-velocity air washers allows them to be used in smaller equipment than those operating with lower air velocities. High capacities per unit of space available from high-velocity spray devices allow practical prefabrication of central station units in either completely assembled and transportable form or, for large-capacity units, easily handled modules. Manufacturers supply units with capacities of up to 150,000 cfm shipped in one piece, including spray system, eliminators, pump, fan, dampers, filters, and other functional components. Such units are self-housed, prewired, prepped, and ready for hoisting into place. The number and arrangement of nozzles vary with different capacities and manufacturers. Adequate values of saturation effectiveness and heat transfer effectiveness are achieved by using higher spray density. Eliminator blades come in varying shapes, but most are a series of aerodynamically clean, sinusoidal shapes. Collected moisture flows down grooves or hooks designed into their profiles, then drains into the storage tank. Washers may be built with shallow drain pans and connected to a central storage tank. High-velocity washers are rectangular in cross section and, except for the eliminators, are similar in appearance and construction to conventional lower-velocity types. Pressure loss is in the range of 0.5 to 1.5 in. of water. These washers are available either as freestanding separate devices for incorporation into field-built central stations or in complete preassembled central station packages from the factory.

5. HUMIDIFICATION/DEHUMIDIFICATION

Air can be humidified with air washers and rigid media by (1) using recirculated water without prior heating of the air, (2) preheating the air and humidifying it with recirculated water, or (3) preheating recirculated water. Precise humidity control may be achieved by arranging rigid media in one or more banks in depth, height, or width, or by providing a controlled bypass. Each bank is activated independently of the others to achieve the desired humidity. In any evaporative humidification application, air should not be allowed to enter the process with a wet-bulb temperature of less than 39°F, or the water may freeze.

Recirculation Without Preheating. Except for the small amount of energy added by the recirculating pump and the small amount of heat leakage into or from the apparatus (including the pump and its connecting piping), the process is adiabatic. Water temperature in the collection basin closely approaches the thermodynamic wet-bulb temperature of the entering air, but it cannot be brought to complete saturation. The psychrometric state point of the leaving air is on the constant thermodynamic wet-bulb temperature line with its end state determined by the saturation effectiveness of the device. Leaving humidity conditions may be controlled using the saturation effectiveness of the process by bypassing air around the evaporative process.

Preheating Air. Preheating air entering an evaporative humidifier increases both the dry- and wet-bulb temperatures and lowers the relative humidity, but it does not alter the air's humidity ratio (mass ratio of water vapor to dry air). As a result, preheating allows more water to be absorbed per unit mass of the same saturation effectiveness. Control is achieved by varying the amount of air preheating at a constant saturation effectiveness. Control precision is a direct function of saturation effectiveness, and a high degree of correlation may be achieved between leaving air and leaving dew-point temperatures when high-saturation-effectiveness devices are used.

Heated Recirculated Water. If heat is added to the water, the process state point of the mixture moves toward the temperature of the heated water (Figure 9A). Elevating the water temperature makes it possible to raise the air dry- and wet-bulb temperatures above the dry-bulb temperature of the entering air with the leaving air becoming fully saturated. Relative humidity of the leaving air can be controlled by (1) bypassing some of the air around the media banks and remixing the two airstreams downstream by using dampers or (2) by automatically reducing the number of operating media banks through pump staging or by operating valves in the different distribution branches.

The following table shows the saturation or humidifying effectiveness of a spray air washer for various spray arrangements. The degree of saturation depends on the extent of contact between air and water. Other conditions being equal, a low-velocity airflow is conducive to higher humidifying effectiveness.

Bank Arrangement	Length, ft	Effectiveness, %
1 downstream	4	50 to 60
	6	60 to 75
1 upstream	6	65 to 80
	8 to 10	80 to 90
2 downstream	8 to 10	85 to 95
2 opposing	8 to 10	85 to 95
2 upstream	8 to 10	90 to 98

Dehumidification with Air Washers and Rigid Media

Air washers and rigid-media direct evaporative coolers may also be used to cool and dehumidify air. Compared to a typical chilled-water or direct-expansion (DX) cooling coil, direct-contact dehumidification can significantly reduce fan power requirements, static pressure losses, and energy consumption (El-Morsi et al. 2003). As shown in Figure 9B, heat and moisture removed from the air raise the water temperature. If the entering water temperature is below the entering air dew point, both the dry- and wet-bulb temperatures of the air are reduced, resulting in cooling and dehumidification. The vapor pressure difference between the entering air and water cools the air. Moisture is transferred from the air to the water, and condensation occurs. Air leaving an evaporative dehumidifier is typically saturated, usually with less than 1°F difference between leaving dry- and wet-bulb temperatures. The difference between the leaving air and water temperatures depends on the difference between entering dry- and wet-bulb temperatures and the process effectiveness, which may be affected by factors such as length and height of the spray chamber, air velocity, quantity of water flow, and spray pattern. Final water conditions are typically 1 to 2°F below the leaving air temperature, depending on the saturation effectiveness of the device used. The common design value for the water temperature rise is usually between 6 and 12°F for refrigerant-chilled water and normal air-conditioning applications, although higher rises are possible and have been used successfully. A smaller rise may be considered when water is chilled by mechanical refrigeration. If warmer water is used, less mechanical refrigeration is required; however, a larger quantity of chilled water is needed to do the same amount of sensible cooling. An economic analysis may be required to determine the best alternative. For humidifiers receiving water from a thermal storage or other low-temperature system, a design with a high temperature rise and minimum water flow may be desirable.

Performance Factors. An evaporative dehumidifier has a performance factor of 1.0 if it can cool and dehumidify the entering air to a wet-bulb temperature equal to the leaving water temperature. This represents a theoretical maximum value that is thermodynamically impossible to achieve. Performance is maximized when both water surface area and air/water contact is maximized. The actual performance factor F_p of any evaporative dehumidifier is less than one and is calculated by dividing the actual air enthalpy change by the theoretical maximum air enthalpy change where

$$F_p = \frac{h_1 - h_2}{h_1 - h_3} \tag{3}$$

where
 h_1 = enthalpy at wet-bulb temperature of entering air, Btu/lb
 h_2 = enthalpy at wet-bulb temperature of leaving air at actual condition, Btu/lb
 h_3 = enthalpy of air at wet-bulb temperature leaving a dehumidifier with $F_p = 1.0$, Btu/lb

Air Cleaning

Air washers and rigid-media direct evaporative cooling equipment can remove particulate and gaseous contaminants with varying degrees of effectiveness through wet scrubbing (which is discussed in Chapter 30). Particle removal efficiencies of rigid media and air washers differ due to differences in equipment construction and principles of operation. Removal also depends largely on the size, density, wettability, and/or solubility of the contaminants to be removed. Large, wettable particles are the easiest to remove. The primary mechanism of separation is by impingement of particles on a wetted surface, which includes eliminator plates in air washers and corrugations of wetted rigid media. Spraying is relatively ineffective in removing most atmospheric dusts. Because the force of impact increases with the size of the solid, the impact (together with the adhesive quality of the wetted surface) determines the device's usefulness as a dust remover. In practice, air-cleaning results of air washers and rigid-media direct-evaporative coolers are typical of comparable impingement filters. Air washers are of little use in removing soot particles because of the lack of adhesion to a greasy surface. They are also relatively ineffective in removing smoke because the particles are too small (less than 1 μm) to impact and be retained on the wet surfaces. Despite their air-cleaning performance, rigid media should not be used for primary filtering. When a rigid-media cooler is placed in an unfiltered airstream, it can quickly become fouled with airborne dust and fibrous debris. When wet, debris can collect in the recirculation basin and in the media, feeding bacterial growth. Bacteria in the air can propagate in waste materials and debris and cause microbial slimes. Filtering entering air is the most effective way to keep debris from accumulating in rigid media. With high-efficiency filters upstream from the cells, most microbial agents and nutrients can be removed from the airstream. Replace rigid media if the corrugations are filled with contaminants when they are dry.

6. SOUND ATTENUATION

Although evaporative cooling media pads are not intended for use as a sound attenuator in air-conditioning systems, tests by Munters Corp. (2002) have shown significant insertion loss, especially in the higher-frequency octave-band center frequency range of 4000 and 8000 Hz. This is of special interest in noise-sensitive applications such as gas turbine inlet cooling systems. Different depths of evaporative cooling media were tested at different face velocities at both wet- and dry-pad conditions. Net insertion loss in the third band (250 Hz) ranged between 2 and 3 dB for both 12 and 16 in. deep media at measured face velocities of 400 to 750 fpm.

7. MAINTENANCE AND WATER TREATMENT

Regular inspection and maintenance of evaporative coolers, air washers, and ancillary equipment ensures proper service and efficiency. Manufacturers' recommendations for maintenance and operation should be followed to help ensure safe, efficient operation. Water lines, water distribution troughs or sumps, pumps, and pump filters must be clean and free of dirt, scale, and debris. They must be constructed so that they can be easily flushed and cleaned. Inadequate water flow causes dry areas on the evaporative media, which reduces the saturation effectiveness and useful life. Motors and bearings should be lubricated and fan drives checked periodically. Water and air filters should be cleaned or replaced as required. The sump water level must be kept below the bottom of the pads, yet high enough to prevent air from short-circuiting below the pads. Bleeding off some water is the most practical means to minimize scale accumulation. The bleed rate should be 5 to 100% of the evaporation rate, depending on water hardness and airborne contaminant level. The water circulation pump should be used to bleed off water (suction by a draw-through fan will otherwise prevent the bleed system from operating effectively). A flush-out cycle, which runs fresh water through the pad every 24 h when the fan is off, may also be used. This water should run for 3 min for every foot of media height. Steam or high-temperature water should not be used to clean either heat pipes or thermosiphon loop air-to-air heat exchangers. High pressure generated by the refrigerant charge inside the individual tubes can severely damage the heat pipes and thermosiphons. Regular inspections should be made to ensure that the bleed rate is adequate and is maintained. Some manufacturers provide a purge cycle in which the entire sump is purged of water and accumulated debris. This cycle helps maintain a cleaner system and may actually save water compared to a standard bleed system. Purge frequency depends on water quality as well as the amount and type of external contaminants. Sumps should have drain couplings on the bottom rather than on the side, to drain the sump completely. Additionally, the sump bottom should slope toward the drain (approximately 0.25 in. per foot of sump length) to facilitate complete draining.

Water Treatment. An effective water treatment and biocide program for cooling towers is not necessarily good practice for evaporative coolers. Evaporative coolers and cooling towers differ significantly: evaporative coolers are directly connected with the supply airstream, whereas cooling towers only indirectly affect the supply air. The effect a biocide may have on evaporative media (both direct and indirect systems) as well as the potential for offensive and/or harmful residual off-gassing must be considered. Pretreatment of a water supply with chemicals intended to hold dissolved material in suspension is best prescribed by a water treatment specialist. Water treated by a zeolite ion exchange softener should not be used because the zeolite exchange of calcium for sodium results in a soft, voluminous scale that may cause dust problems downstream. Any chemical agents used should not promote microbial growth or harm the cabinet, media, or heat exchanger materials. This topic is discussed in more detail in Chapter 49 of the 2015 ASHRAE Handbook—HVAC Applications. Consider the following factors for water treatment:

- Use caution when using very pure water from reverse osmosis or deionization in media-based evaporative coolers. This water does not wet random media well, and it can deteriorate many types of media because of its corrosive nature. The same problem can occur in a once-through water distribution system if the water is very pure.
- Periodically check for algae, slime, and bacterial growth. If required, add a biocide registered for use in evaporative coolers by an appropriate agency, such as the U.S. Environmental Protection Agency (EPA).

Ozone-generation systems have been used as an alternative to standard chemical biocide water treatments. Ozone can be produced on site (eliminating chemical storage) and injected into the water circulation system. It is a fast-acting oxidizer that rapidly breaks down to nontoxic compounds. In low concentrations, ozone is benign to humans and to the materials used in evaporative coolers. Algae can be minimized by reducing the media and sump exposure to nutrient and light sources (by using hoods, covers, and prefilters), by keeping the bottom of the media out of standing water in the sump, and by allowing the media to completely dry out every 24 h. **Scale.** Units that have heat exchangers with a totally wetted surface and materials that are not harmed by chemicals can be descaled periodically with a commercial descaling agent and then flushed out. Mineral scale deposits on a wetted indirect evaporative heat exchanger are usually soft and allow wetting through to and evaporation at the surface of the heat exchanger. Excess scale thickness reduces heat transfer and should be removed. **Nonchemical Water Treatment.** Makeup and recirculation water furnished to a rigid-media-pad, direct evaporative cooler should be treated to reduce the risk of airborne microbial or particulate contamination of the building supply air. See Chapter 49 of the 2015 ASHRAE Handbook—HVAC Applications for more information on nonchemical water treatment. **Air Washers.** The air washer spray system requires the most attention. Partially clogged nozzles are indicated by a rise in spray pressure; a fall in pressure is symptomatic of eroded orifices. Strainers can minimize this problem. Continuous operation requires either a bypass around pipeline strainers or duplex strainers. Air washer tanks should be drained and dirt deposits removed regularly. Eliminators and baffles should be periodically inspected and repainted to prevent corrosion damage. **Freeze Protection.** In colder climates, evaporative coolers must be protected from freezing. This is usually done seasonally by simply draining the cooler and the water supply line with solenoid valves. Often, an outdoor air temperature sensor initiates this action. It is important that drain solenoid valves be of zero-differential design. If a heat exchanger coil is used, the tubes must be horizontal so they will drain to the lowest part of their manifold.

Legionnaires' Disease

Legionnaires' disease is contracted by inhaling into the lower respiratory system an aerosol (1 to 5 μm in diameter) laden with sufficient *Legionella pneumophila* bacteria. Evaporative coolers do not provide suitable growth conditions for the bacteria and generally do not release an aerosol. A good maintenance program eliminates potential microbial problems and reduces the concern for disease transmission (ASHRAE 2000, 2015; Puckorius et al. 1995). There have been no known cases of Legionnaires' disease in air washers or wetted-media evaporative coolers/humidifiers, and there is no positive association of Legionnaires' disease with indirect evaporative coolers (ASHRAE Guideline 12-2000). The following precautions and maintenance procedures for water systems also improve cooler performance, reduce microbial growth and musty odors, and prolong equipment life:

- Run fans after turning off water until the media completely dries.
- Thoroughly clean and flush the entire cooling water loop regularly (minimum monthly). Disinfect before and after cleaning.
- Avoid dead-end piping, low spots, and other areas in the water distribution system where water may stagnate during shutdown.
- Obtain and maintain the best available mist elimination technology, especially when using misters and air washers.
- Do not locate the evaporative cooler inlet near a cooling tower outlet.
- Maintain system bleedoff and/or purge consistent with makeup water quality.
- Maintain system cleanliness. Deposits from calcium carbonate, minerals, and nutrients may contribute to growth of molds, slime, and other microbes annoying to building occupants.
- Develop a maintenance checklist, and follow it on a regular basis.
- Consult the equipment or media manufacturer for more detailed assistance in water system maintenance and treatment.

ASHRAE Standard 188-2015 gives detailed guidance on risk management for building water systems.

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