CHAPTER 40

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. Residential and industrial humidification equipment are covered in Chapter 21.

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
- Provide humidification and dehumidification when needed
- Easy to use with direct digital control (DDC)
- Reduced pollution emissions
- No chlorofluorocarbon (CFC) usage

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 air-cooling equipment.

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. 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).

In **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 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.

Direct and indirect evaporative processes can be **combined** (**indirect/direct**). The first stage (indirect) sensibly cools the air, which is then passed through the second stage (direct) and evaporatively cooled further. Combination systems use both direct and indirect evaporative principles as well as secondary heat exchangers and cooling coils. Secondary heat exchangers enhance both cooling and heat recovery (in winter), and the coils provide additional cooling/ dehumidification as needed. Used in both dual-duct and unitary systems, secondary heat exchangers can also save energy by eliminating the need for terminal reheat in some applications (in such systems, air may exit below the initial wet-bulb temperature).

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).

DIRECT EVAPORATIVE AIR COOLERS

In direct evaporative air cooling, air is drawn through porous wetted pads or a spray and its sensible heat energy evaporates some water; the heat and mass transfer between the air and water lowers the air dry-bulb temperature and increases the humidity at a constant wet-bulb temperature. The dry-bulb temperature of the nearly saturated air approaches the ambient air's wet-bulb temperature. The process is adiabatic, so no sensible cooling occurs.

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, or the extent to which complete saturation is approached, is expressed as the **direct saturation effectiveness**, which is defined as:

$$\varepsilon_e = 100\frac{t_1 - t_2}{t_1 - t_s'} \qquad (1)$$

where

ε_e = direct evaporative cooling or saturation effectiveness, %
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 a high saturation effectiveness) 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 insufficient for indoor comfort when the ambient wet-bulb temperature is higher than about 70°F; however, cooling is still sufficient for relief cooling applications (e.g., greenhouses, industrial cooling). Direct evaporative coolers should not recirculate indoor air.

The preparation of this chapter is assigned to TC 5.7, Evaporative Cooling.

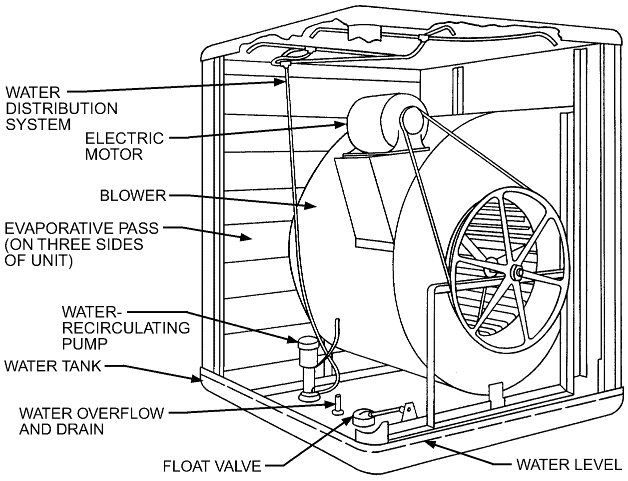

Fig. 1 Typical Random-Media Evaporative Cooler

Random-Media Air Coolers

These coolers contain evaporative pads, usually of aspen wood or absorbent plastic fiber/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 forces air through the evaporative pads and delivers it to the space to be cooled. 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 18000 cfm). Cooler selection should be based on a capacity rating from an independent agency.

When clean and well maintained, commercial random-media air coolers operate at approximately 80% effectiveness and remove 10 μm and larger particles from the air. In some units, supplementary filters before or after the evaporative pads keep particles from entering the cooler, even when it is operated without water to circulate fresh air. 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 coolers are usually designed for an evaporative pad face velocity of 100 to 250 fpm, with a pressure drop of 0.1 in. of water. Aspen fibers are packed to approximately 0.3 to 0.4 lb/ft^2 of face area based on a 2 in. thick pad. Pads are mounted 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 pumps water through tubes that provide an equal flow of water to each trough. It is important that pumps are thermally protected. The sump or water tank has a water makeup connection, float valve, overflow pipe, and drain. Provisions to bleed water to prevent the buildup of minerals, dirt, and microbial growth are typically incorporated in the design.

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 that have been treated to absorb water yet resist its 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. The medium has the desirable characteristics of low resistance to airflow, high saturation effectiveness, and self-cleaning by flushing the front face of the pad. The rigid medium is usually designed for a face velocity of 400 to 600 fpm.

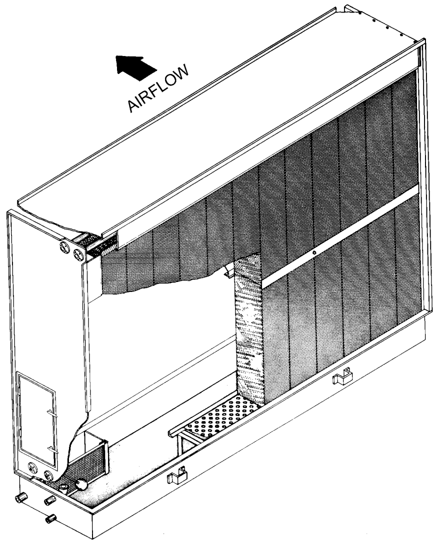

Fig. 2 Typical Rigid-Media Air Cooler

Direct evaporative air coolers using this material are built to handle as much as 600,000 cfm with or without fans. 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 and clean rooms), or a pump may recirculate the water from a lower reservoir constructed of heavy-gage 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.

Flanges at the entering and leaving faces allow the unit to be connected to 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 because 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 is evaporatively cooled as it is drawn through pads located on the other end of the building. The pads are wetted from above by a perforated pipe, and excess water is collected for recirculation. In some cases, the pads are wetted with high-pressure fogging nozzles, which provide additional cooling. Water from fogging nozzles must never be recirculated. The pad should be sized for 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.

INDIRECT EVAPORATIVE AIR COOLERS

Packaged Indirect Evaporative Air Coolers

An indirect evaporative cooling (IEC) heat exchanger is illustrated in Figure 3. This cross-flow, plate-type heat exchanger uses a recirculation sump water pump to wet the inside of the heat

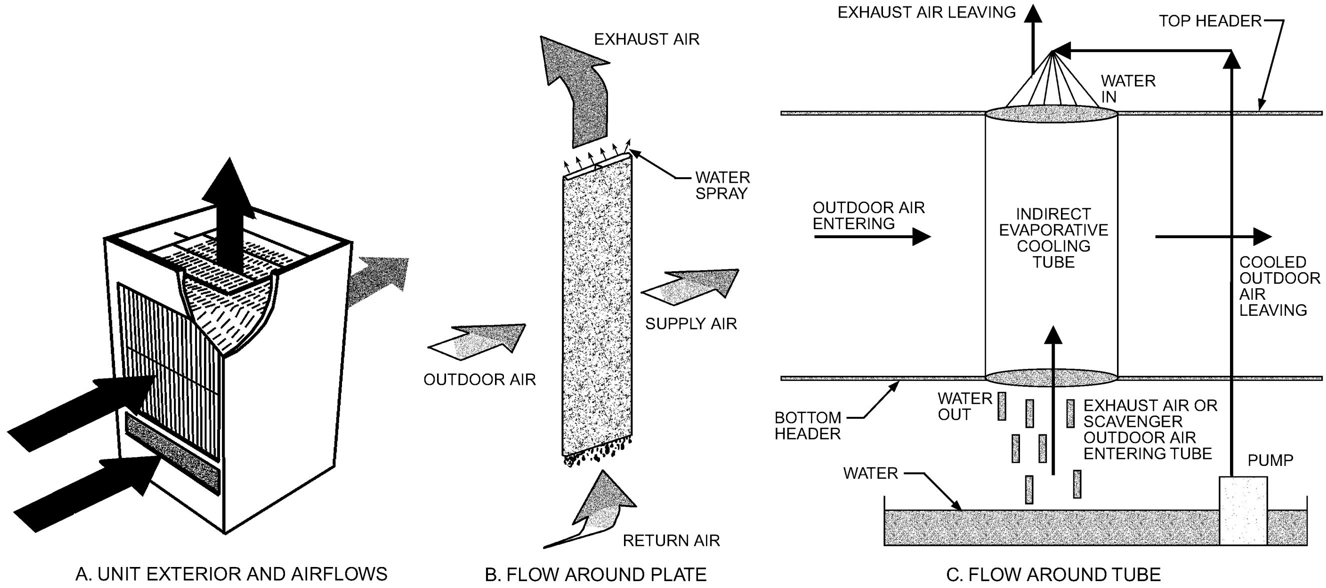

Fig. 3 Indirect Evaporative Cooling (IEC) Heat Exchanger
(Courtesy Munters/Des Champs)

exchanger tubes. Either building return or outdoor air may be drawn up through the inside of the tubes by a secondary air fan. Outdoor air entering the building is sensibly cooled by the exterior surface of the tubes, which are chilled by water evaporating off their interior surface. Latent cooling may also occur if the secondary air wet-bulb temperature is below the outdoor air dew point.

These heat exchangers are capable of a 60 to 80% approach of the ambient dry-bulb temperature to the secondary airflow entering wet-bulb temperature. This is called **wet-bulb depression efficiency (WBDE)**, and is calculated as follows:

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

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 air flow penalties range from 0.8 to 0.9 in. of water. Secondary airflow ratios are selected in the range of 1 to 1 down to a low of 1 cfm of outdoor air (OA) to 0.7 cfm of secondary airflow. Cooling energy efficiency ratios (EER) for this type of heat exchanger range from 40 to 80.

With DX Refrigeration. Figure 4 illustrates a package unit design that combines the plate-type indirect evaporative cooling heat exchanger with a direct-expansion (DX) refrigeration final stage of cooling. The geometry of the plate-type heat exchanger usually limits the size of this application to less than 20,000 cfm of supply air.

By placing the condenser coil in the wet-side air path off the heat exchanger, the mechanical cooling component's coefficient of performance (COP) is significantly increased 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. The 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.

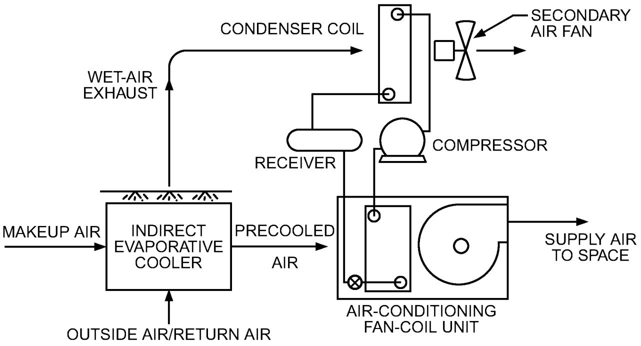

Fig. 4 Indirect Evaporative Cooler Used as Precooler

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 air flows. 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

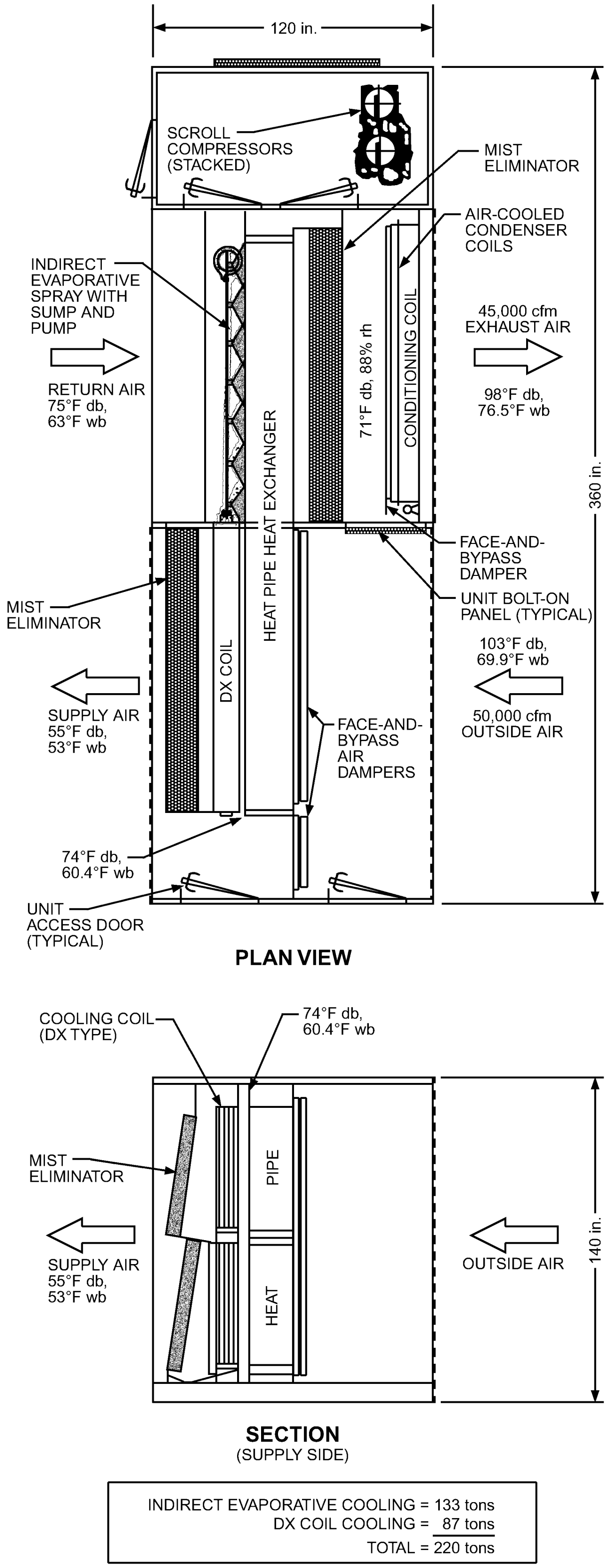

Fig. 5 Heat Pipe Indirect Evaporative Cooling (IEC) Heat Exchanger Packaged with DX System

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 refrigeration 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 are used to control condensing head pressure within an acceptable range. During winter, when the heat pipe heat exchanger is used for heat recovery 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 shown 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 EER of 17.1. A conventional air-cooled condensing unit on the roof in 100°F ambient temperatures would typically require 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 the evaporatively cooled refrigeration design would only consume a total of 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 51 of the 2007 *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 (Scofield 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. For new installations, equipment can be downsized, lowering overall project and operational costs. Chapter 25 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 is pumped 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 is evaporatively cooled 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 is again cooled 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. Also, the tower is more accessible for maintenance. Overall WBDE ε_e may range between 55% and 75% or higher. If return air is sent to the cooling tower of an indirect cooling system before being discharged outside, 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 outside (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 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.

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 the secondary air passes through the direct evaporative cooler, the dry-bulb temperature is lowered by evaporative cooling. As this air passes through the heat wheel, the mass of the medium is cooled 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 secondary air wet-bulb temperature.

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 of refrigeration. 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 an average annual power consumption greater than 1.0 kW/ton.

In dry environments, indirect/direct evaporative cooling is usually designed to supply 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 that should be considered in systems such as the one in Figure 7 include air filters on the entering side of each heat wheel or pipe. Systems without the direct evaporative cooler on the exhaust side have been successfully used in many laboratory applications in the southwestern United States. Maintenance inside a laboratory exhaust airstream can be hazardous; therefore, the fewer components in the airstream needing maintenance, the lower the risk to staff.

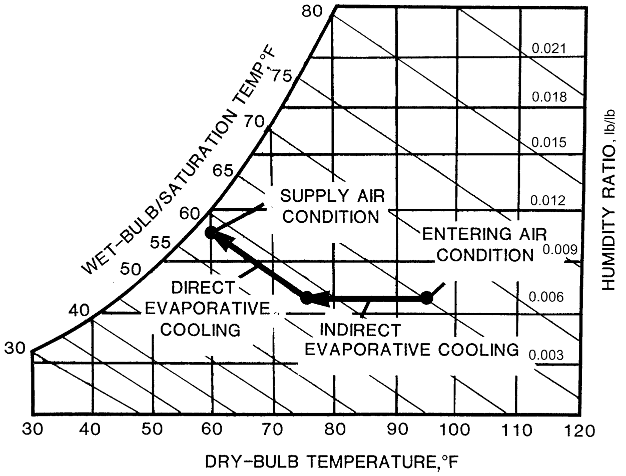

Fig. 6 Combination Indirect/Direct Evaporative Cooling Process

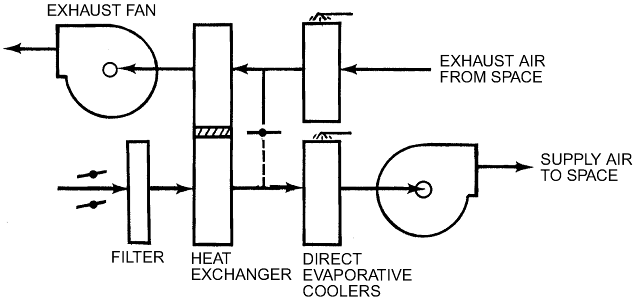

Fig. 7 Indirect/Direct Evaporative Cooler with Heat Exchanger (Rotary Heat Wheel or Heat Pipe)

In areas with a higher 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 is energized 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 excess moisture removal 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.

A single coil may be used to cool return chilled water with cooling tower water and a plate heat exchanger (a form of indirect evaporative cooling). This hybrid, three-stage configuration allows indirect cooling when the wet-bulb temperature is low and mechanical cooling when the wet bulb is high or when dehumidification is necessary.

The designer should consider using building exhaust and/or outside 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 outside air and building exhaust as the secondary airstream; whichever source has the lower wet-bulb temperature would 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 outside air. In this case, outside 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 outside air; mixing of airstreams; bypass of components, or variable-volume control. Controllable elements include

- Modulating outside 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.

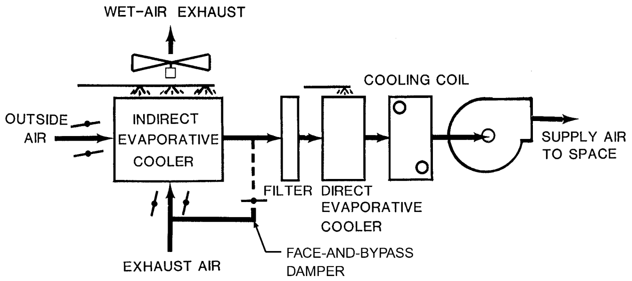

Fig. 8 Three-Stage Indirect/Direct Evaporative Cooler

Precooling and Makeup Air Pretreatment

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

The condenser may be cooled by adding a direct evaporative cooler (usually without a fan) to the condenser fan inlet. 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 from 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 outside air or building exhaust air (whichever has the lower wet-bulb temperature) can be used as the secondary air source. Outside 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, outside air cooling has higher energy savings and lower electrical demand savings than return air cooling. These systems can significantly reduce the outside air load and should be analyzed using a psychrometric process for the region and climate being considered.

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.

The 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.

The 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 permit recirculation of spray water for control purposes in dehumidification work. The excess then returns over the weir to the central water-chilling machine.

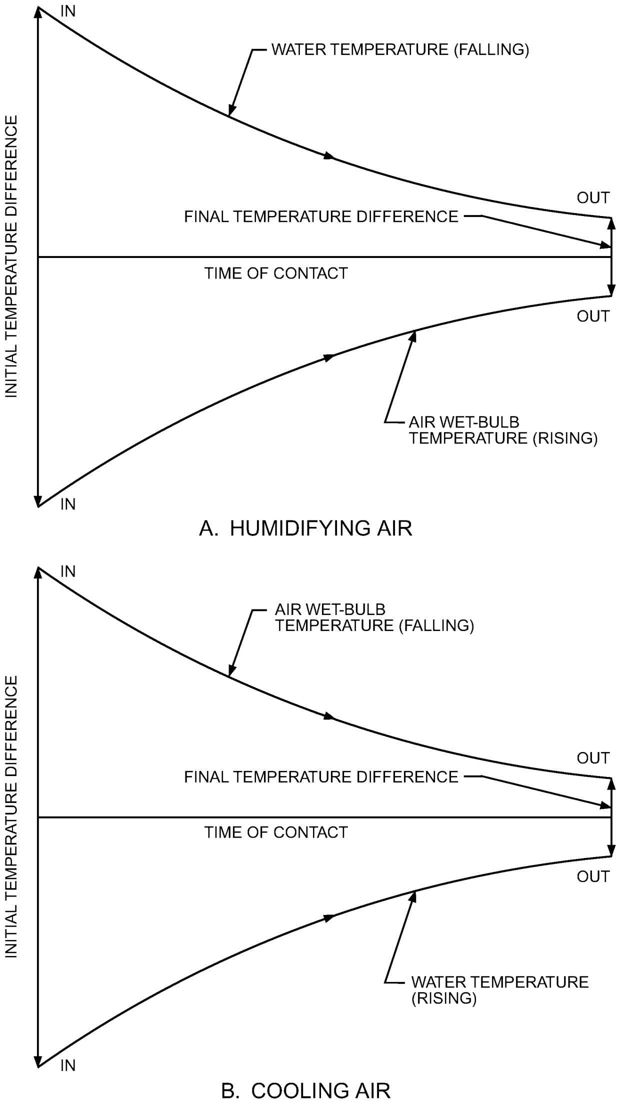

Fig. 9 Interaction of Air and Water in Air Washer Heat Exchanger

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, prepiped, 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.

HUMIDIFICATION/DEHUMIDIFICATION

Humidification with Air Washers and Rigid Media

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 permitted 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 the air entering an evaporative humidifier increases both the dry- and wet-bulb temperatures and lowers the relative humidity, but it does not alter the humidity ratio (mass ratio of water vapor to dry air) of the air. As a result, preheating permits more water to be absorbed per unit mass of dry air passing through the process at 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
2 downstream	8 to 10	80 to 90
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 29). 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.

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. 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 that 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.

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 outside 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 48 of the 2007 *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, louvers 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.

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 outside 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 transmittal (ASHRAE 1998, 2000; Puckorius et al. 1995). There have been no known cases of Legionnaires' disease with 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.

REFERENCES

ASHRAE. 1998. *Legionellosis: Position statement*.

ASHRAE. 2000. *Minimizing the risk of Legionnaires' disease.*

ASHRAE. 2000. Minimizing the risk of legionellosis associated with building water systems. *Guideline* 12-2000.

El-Morsi, M., S.A. Klein, and D.T. Reindl. 2003. Air washers—A new look at a vintage technology. *ASHRAE Journal* 45(10):32-36.

Foster, R.E. and E. Dijkstra. 1996. Evaporative air-conditioning fundamentals: Environmental and economic benefits worldwide. *Refrigeration Science and Technology Proceedings*. International Institute of Refrigeration, Danish Technological Institute, Danish Refrigeration Association, Aarhus, Denmark, pp. 101-110.

Mathur, G.D. 1990. Indirect evaporative cooling with two-phase thermosiphon coil loop heat exchangers. *ASHRAE Transactions* 96(1):1241-1249.

Mathur, G.D. 1991. *Indirect evaporative cooling with heat pipe heat exchangers*. ASME Book No. NE(5):79-85.

Peterson, J.L. 1993. An effectiveness model for indirect evaporative coolers. *ASHRAE Transactions* 99(2):392-399.

Puckorius, P.R., P.T. Thomas, and R.L. Augspurger. 1995. Why evaporative coolers have not caused Legionnaires' disease. *ASHRAE Journal* 37(1): 29-33.

Scofield, M. 1986. The heat pipe used for dry evaporative cooling. *ASHRAE Transactions* 92(1B):371-381.

Scofield, M. and N.H. DesChamps. 1984. Indirect evaporative cooling using plate type heat exchangers. *ASHRAE Transactions* 90(1):148-153.

Woolridge, M.J., H.L. Chapman, and D. Pescod. 1976. Indirect evaporative cooling systems. *ASHRAE Transactions* 82(1):146-155.

BIBLIOGRAPHY

Anderson, W.M. 1986. Three-stage evaporative air conditioning versus conventional mechanical refrigeration. *ASHRAE Transactions* 92(1B):358-370.

Eskra, N. 1980. Indirect/direct evaporative cooling systems. *ASHRAE Journal* 22(5):21-25.

Felver, T., M. Scofield, and K. Dunnavant. 2001. Cooling California's computer centers. *HPAC Engineering* (March):59.

Scofield, M. 1987. Unit gives 45 tons of cooling without a compressor. *Air Conditioning, Heating and Refrigeration News* (December):23.

Scofield, M. and N.H. DesChamps. 1980. EBTR compliance and comfort cooling too! *ASHRAE Journal* 22(6):61-63.

Supple, R.G. 1982. Evaporative cooling for comfort. *ASHRAE Journal* 24(8):36.

Watt, J.R. 1986. *Evaporative air conditioning handbook*. Chapman and Hall, London.