CHAPTER 22

AIR-COOLING AND DEHUMIDIFYING COILS

MOST equipment used today for cooling and dehumidifying an airstream under forced convection incorporates a coil section that contains one or more cooling coils assembled in a coil bank arrangement. Such coil sections are used extensively as components in room terminal units; larger factory-assembled, self-contained air conditioners; central station air handlers; and field built-up systems. Applications of each coil type are limited to the field within which the coil is rated. Other limitations are imposed by code requirements, proper choice of materials for the fluids used, the configuration of the air handler, and economic analysis of the possible alternatives for each installation.

USES FOR COILS

Coils are used for air cooling with or without accompanying dehumidification. Examples of cooling applications without dehumidification are (1) precooling coils that use well water or other relatively high-temperature water to reduce load on the refrigerating equipment and (2) chilled-water coils that remove sensible heat from chemical moisture-absorption apparatus. The heat pipe coil is also used as a supplementary heat exchanger for preconditioning in air-side sensible cooling (see Chapter 25). Most coil sections provide air sensible cooling and dehumidification simultaneously.

The assembly usually includes a means of cleaning air to protect the coil from dirt accumulation and to keep dust and foreign matter out of the conditioned space. Although cooling and dehumidification are their principal functions, cooling coils can also be wetted with water or a hygroscopic liquid to aid in air cleaning, odor absorption, or frost prevention. Coils are also evaporatively cooled with a water spray to improve efficiency or capacity. Chapter 40 has more information on indirect evaporative cooling. For general comfort conditioning, cooling, and dehumidifying, the **extended-surface (finned) cooling coil** design is the most popular and practical.

COIL CONSTRUCTION AND ARRANGEMENT

In finned coils, the external surface of the tubes is primary, and the fin surface is secondary. The primary surface generally consists of rows of round tubes or pipes that may be staggered or placed in line with respect to the airflow. Flattened tubes or tubes with other nonround internal passageways are sometimes used. The inside surface of the tubes is usually smooth and plain, but some coil designs have various forms of internal fins or turbulence promoters (either fabricated or extruded) to enhance performance. The individual tube passes in a coil are usually interconnected by return bends (or hairpin bend tubes) to form the serpentine arrangement of multipass tube circuits. Coils are usually available with different circuit arrangements and combinations offering varying numbers of parallel water flow passes within the tube core (Figure 1).

Cooling coils for water, aqueous glycol, brine, or halocarbon refrigerants usually have aluminum fins on copper tubes, although

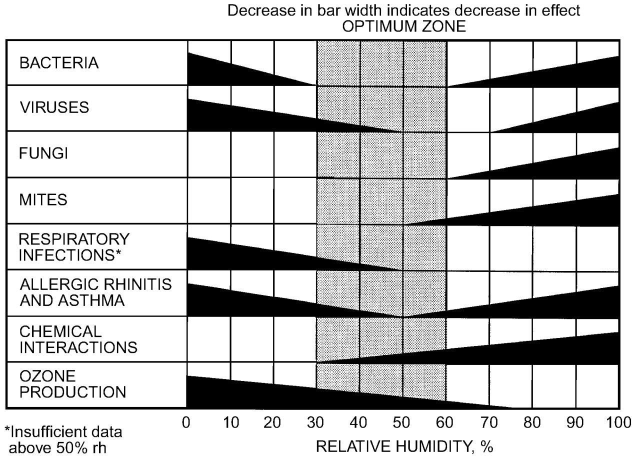

Fig. 1 Typical Water Circuit Arrangements

copper fins on copper tubes and aluminum fins on aluminum tubes (excluding water) are also used. Adhesives are sometimes used to bond header connections, return bends, and fin-tube joints, particularly for aluminum-to-aluminum joints. Certain special-application coils feature an all-aluminum extruded tube-and-fin surface.

Common core tube outside diameters are 5/16, 3/8, 1/2, 5/8, 3/4, and 1 in., with fins spaced 4 to 18 per inch. Tube spacing ranges from 0.6 to 3.0 in. on equilateral (staggered) or rectangular (in-line) centers, depending on the width of individual fins and on other performance considerations. Fins should be spaced according to the job to be performed, with special attention given to air friction; possibility of lint accumulation; and frost accumulation, especially at lower temperatures.

Tube wall thickness and the required use of alloys other than copper are determined mainly by the coil's working pressure and safety factor for hydrostatic burst (pressure). Maximum allowable working pressure (MAWP) for a coil is derived according to ASME's *Boiler and Pressure Vessel Code*, Section VIII, Division 1 and Section II (ASTM material properties and stress tables). Pressure vessel safety standards compliance and certifications of coil construction may be required by regional and local codes before field installation. Fin type and header construction also play a large part in determining wall thickness and material. Local job site codes and applicable nationally recognized safety standards should be consulted in coil design and application.

This type of air-cooling coil normally has a shiny aluminum air-side surface. For special applications, the fin surface may be copper or have a brown or blue-green dip-process coating. These coatings protect the fin from oxidation that occurs when common airborne corrosive contaminants are diluted on a wet (dehumidifying) surface. Corrosion protection is increasingly important as indoor air quality (IAQ) guidelines call for higher percentages of outside air. Baked-on or anodized coating improves the expected service life

The preparation of this chapter is assigned to TC 8.4, Air-to-Refrigerant Heat Transfer Equipment.

compared to plain aluminum fins under similar conditions. Uncoated fins on non-dehumidifying, dry cooling coils are generally not affected by normal ambient airborne chemicals, except, to some extent, in a saline atmosphere. Once the coil is installed, little can be done to improve air-side protection.

Incoming airstream stratification across the coil face reduces coil performance. Proper air distribution is defined as having a measured airflow anywhere on the coil face that does not vary more than 20%. Moisture carryover at the coil's air leaving side or uneven air filter loading are indications of uneven airflow through the coil. Normal corrective procedure is to install inlet air straighteners, or an air blender if several airstreams converge at the coil inlet face. Additionally, condensate water should never be allowed to saturate the duct liner or stand in the drain pan (trough). The coil frame (particularly its bottom sheet metal member) should not be allowed to sit in a pool of water, to prevent rusting.

Water and Aqueous Glycol Coils

Good performance of water-type coils requires both eliminating all air and water traps in the water circuit and the proper distribution of water. Unless properly vented, air may accumulate in the coil tube circuits, reducing thermal performance and possibly causing noise or vibration in the piping system. Air vent and drain connections are usually provided on coil water headers, but this does not eliminate the need to install, operate, and maintain the coil tube core in a level position. Individual coil vents and drain plugs are often incorporated on the headers (Figure 1). Water traps in tubing of a properly leveled coil are usually caused by (1) improper nondraining circuit design and/or (2) center-of-coil downward sag. Such a situation may cause tube failure (e.g., freeze-up in cold climates or tube erosion because of untreated mineralized water).

Depending on performance requirements, fluid velocity inside the tube usually ranges from approximately 1 to 8 fps for water and 0.5 to 6 fps for glycol. When turbulators or grooved tubes are used, in-tube velocities should not exceed 4 fps. The design fluid pressure pressure drop across the coils varies from about 5 to 50 ft of water head. For nuclear HVAC applications, ASME *Standard* AG-1, Code on Nuclear Air and Gas Treatment, requires a minimum tube velocity of 2 fps. ARI *Standard* 410 requires a minimum of 1 fps or a Reynolds number of 3100 or greater. This yields more predictable performance.

In certain cases, the water may contain considerable sand and other foreign matter (e.g., precooling coils using well water, or where minerals in the cooling water deposit on and foul the tube surface). It is best to filter out such sediment. Some coil manufacturers offer removable water header plates or a removable plug for each tube that allows the tube to be cleaned, ensuring continuing rated performance while the cooling units are in service. Where build-up of scale deposits or fouling of the water-side surface is expected, a scale factor is sometimes included when calculating thermal performance of the coils. Cupronickel, red brass, bronze, and other tube alloys help protect against corrosion and erosion deterioration caused primarily by internal fluid flow abrasive sediment. The core tubes of properly designed and installed coils should feature circuits that (1) have equally developed line length, (2) are self-draining by gravity during the coil's off cycle, (3) have the minimum pressure drop to aid water distribution from the supply header without requiring excessive pumping head, and (4) have equal feed and return by the supply and return header. Design for proper in-tube water velocity determines the circuitry style required. Multirow coils are usually circuited to the cross-counterflow arrangement and oriented for top-outlet/bottom-feed connection.

Direct-Expansion Coils

Coils for halocarbon refrigerants present more complex cooling fluid distribution problems than do water or brine coils. The coil should cool effectively and uniformly throughout, with even refrigerant distribution. Halocarbon coils are used on two types of refrigerated systems: flooded and direct-expansion.

A flooded system is used mainly when a small temperature difference between the air and refrigerant is desired. Chapter 3 of the 2006 *ASHRAE Handbook—Refrigeration* describes flooded systems in more detail.

For direct-expansion systems, two of the most commonly used refrigerant liquid metering arrangements are the capillary tube assembly (or restrictor orifice) and the thermostatic expansion valve (TXV) device. The **capillary tube** is applied in factory-assembled, self-contained air conditioners up to approximately 10 ton capacity, but is most widely used on smaller-capacity models such as window or room units. In this system, the bore and length of a capillary tube are sized so that at full load, under design conditions, just enough liquid refrigerant to be evaporated completely is metered from the condenser to the evaporator coil. Although this type of metering arrangement does not operate over a wide range of conditions as efficiently as a TXV system, its performance is targeted for a specific design condition.

A **thermostatic expansion valve** system is commonly used for all direct-expansion coil applications described in this chapter, particularly field-assembled coil sections and those used in central air-handling units and larger, factory-assembled hermetic air conditioners. This system depends on the TXV to automatically regulate the rate of refrigerant liquid flow to the coil in direct proportion to the evaporation rate of refrigerant liquid in the coil, thereby maintaining optimum performance over a wide range of conditions. Superheat at the coil suction outlet is continually maintained within the usual predetermined limits of 6 to 10°F. Because the TXV responds to the superheat at the coil outlet, superheat within the coil is produced with the least possible sacrifice of active evaporating surface.

The length of each coil's refrigerant circuits, from the TXV's distributor feed tubes through the suction header, should be equal. The length of each circuit should be optimized to provide good heat transfer, good oil return, and a complementary pressure drop across the circuit. The coil should be installed level, and coil circuitry should be designed to self-drain by gravity toward the suction header connection. This is especially important on systems with unloaders or variable-speed-drive compressor(s). When non-self-drain circuitry is used, the circuit and suction connection should be designed for a minimum tube velocity sufficient to avoid compressor lube oil trapping in the coil.

To ensure reasonably uniform refrigerant distribution in multicircuit coils, a distributor is placed between the TXV and coil inlets to divide refrigerant equally among the coil circuits. The refrigerant distributor must be effective in distributing both liquid and vapor because refrigerant entering the coil is usually a mixture of the two, although mainly liquid by weight. Distributors can be placed either vertically or horizontally; however, the vertical down position usually distributes refrigerant between coil circuits better than the horizontal for varying load conditions.

Individual coil circuit connections from the refrigerant distributor to the coil inlet are made of small-diameter tubing; the connections are all the same length and diameter so that the same flow occurs between each refrigerant distributor tube and each coil circuit. To approximate uniform refrigerant distribution, refrigerant should flow to each refrigerant distributor circuit in proportion to the load on that coil. The heat load must be distributed equally to each refrigerant circuit for optimum coil performance. If the coil load cannot be distributed uniformly, the coil should be recircuited and connected with more than one TXV to feed the circuits (individual suction may also help). In this way, refrigerant distribution is reduced in proportion to the number of distributors to have less effect on overall coil performance when design must accommodate some unequal circuit loading. Unequal circuit loading may also be caused by uneven air velocity across the coil's face, uneven entering

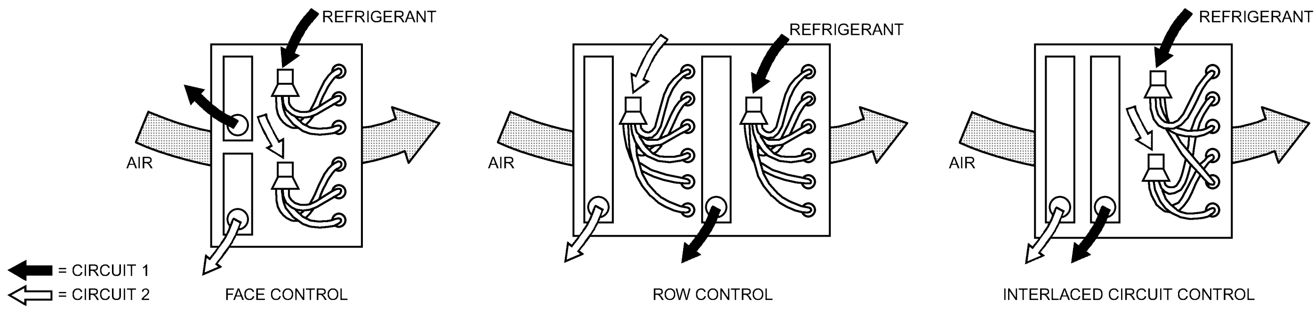

Fig. 2 Arrangements for Coils with Multiple Thermostatic Expansion Valves

air temperature, improper coil circuiting, oversized orifice in distributor, or the TXV's not being directly connected (close-coupled) to the distributor.

Control of Coils

Cooling capacity of water coils is controlled by varying either water flow or airflow. Water flow can be controlled by a three-way mixing, modulating, and/or throttling valve. For airflow control, face and bypass dampers are used. When cooling demand decreases, the coil face damper starts to close, and the bypass damper opens. In some cases, airflow is varied by controlling fan capacity with speed controls, inlet vanes, or discharge dampers.

Chapter 46 of the 2007 *ASHRAE Handbook—HVAC Applications* addresses air-cooling coil control to meet system or space requirements and factors to consider when sizing automatic valves for water coils. Selection and application of refrigerant flow control devices (e.g., thermostatic expansion valves, capillary tube types, constant-pressure expansion valves, evaporator pressure regulators, suction-pressure regulators, solenoid valves) as used with direct-expansion coils are discussed in Chapter 44 of the 2006 *ASHRAE Handbook—Refrigeration*.

For factory-assembled, self-contained packaged systems or field-assembled systems using direct-expansion coils equipped with TXVs, a single valve is sometimes used for each coil; in other cases, two or more valves are used. The thermostatic expansion valve controls the refrigerant flow rate through the coil circuits so refrigerant vapor at the coil outlet is superheated properly. Superheat is obtained with suitable coil design and proper valve selection. Unlike water flow control valves, standard pressure/temperature thermostatic expansion valves alone do not control the refrigeration system's capacity or the temperature of the leaving air, nor do they maintain ambient conditions in specific spaces. However, some electronically controlled TXVs have these attributes.

To match refrigeration load requirements for the conditioned space to the cooling capacity of the coil(s), a thermostat located in the conditioned space(s) or in the return air temporarily interrupts refrigerant flow to the direct-expansion cooling coils by stopping the compressor(s) and/or closing the solenoid liquid-line valve(s). Other solenoids unload compressors by suction control. For jobs with only a single zone of conditioned space, the compressor's on-off control is frequently used to modulate coil capacity. Selection and application of evaporator pressure regulators and similar regulators that are temperature-operated and respond to the temperature of conditioned air are covered in Chapter 44 of the 2006 *ASHRAE Handbook—Refrigeration*.

Applications with multiple zones of conditioned space often use solenoid liquid-line valves to vary coil capacity. These valves should be used where thermostatic expansion valves feed certain types (or sections) of evaporator coils that may, according to load variations, require a temporary but positive interruption of refrigerant flow. This applies particularly to multiple evaporator coils in a unit where one or more must be shut off temporarily to regulate its zone capacity. In such cases, a solenoid valve should be installed directly upstream of the thermostatic expansion valve(s). If more than one expansion valve feeds a particular zone coil, they may all be controlled by a single solenoid valve.

For a coil controlled by multiple refrigerant expansion valves, there are three arrangements: (1) face control, in which the coil is divided across its face; (2) row control; and (3) interlaced circuitry (Figure 2).

Face control, which is the most widely used because of its simplicity, equally loads all refrigerant circuits in the coil. Face control has the disadvantage of permitting condensate reevaporation on the coil portion not in operation and bypassing air into the conditioned space during partial-load conditions, when some of the TXVs are off. However, while the bottom portion of the coil is cooling, some of the advantages of single-zone humidity control can be achieved with air bypasses through the inactive top portion.

Row control, seldom available as standard equipment, eliminates air bypassing during partial-load operation and minimizes condensate reevaporation. Close attention is required for accurate calculation of row-depth capacity, circuit design, and TXV sizing.

Interlaced circuit control uses the whole face area and depth of coil when some expansion valves are shut off. Without a corresponding drop in airflow, modulating refrigerant flow to an interlaced coil increases coil surface temperature, thereby necessitating compressor protection (e.g., suction pressure regulators or compressor multiplexing).

Flow Arrangement

In air conditioning, the relation of the fluid flow arrangement in the coil tubes to coil depth greatly influences performance of the heat transfer surface. Generally, air-cooling and dehumidifying coils are multirow and circuited for **counterflow** arrangement. Inlet air is applied at right angles to the coil's tube face (coil height), which is also at the coil's outlet header location. Air exits at the opposite face (side) of the coil where the corresponding inlet header is located. Counterflow can produce the highest possible heat exchange in the shortest possible (coil row) depth because it has the closest temperature relationships between tube fluid and air at each (air) side of the coil; the temperature of the entering air more closely approaches the temperature of the leaving fluid than the temperature of the leaving air approaches the temperature of the entry fluid. The potential of realizing the highest possible mean temperature difference is thus arranged for optimum performance.

Most direct-expansion coils also follow this general scheme of thermal counterflow, but proper superheat control may require a hybrid combination of parallel flow and counterflow. (Air flows in the same direction as the refrigerant in parallel-flow operation.) Often, the optimum design for large coils is parallel flow in the coil's initial (entry) boiling region followed by counterflow in the superheat (exit) region. Such a hybrid arrangement is commonly used for process applications that require a low temperature difference (low TD).

Coil hand refers to either the right hand (RH) or left hand (LH) for counterflow arrangement of a multirow counterflow coil. There is no convention for what constitutes LH or RH, so manufacturers usually establish a convention for their own coils. Most manufacturers designate the location of the inlet water header or refrigerant distributor as the coil hand reference point. Figure 3 illustrates the more widely accepted coil hand designation for multirow water or refrigerant coils.

Applications

Figure 4 shows a typical arrangement of coils in a field built-up central station system. All air should be filtered to prevent dirt, insects, and foreign matter from accumulating on the coils. The cooling coil (and humidifier, when used) should include a drain pan under each coil to catch condensate formed during cooling (and excess water from the humidifier). The drain connection should be downstream of the coils, be of ample size, have accessible cleanouts, and discharge to an indirect waste or storm sewer. The drain also requires a deep-seal trap so that no sewer gas can enter the system. Precautions must be taken if there is a possibility that the drain might freeze. The drain pan, unit casing, and water piping should be insulated to prevent sweating.

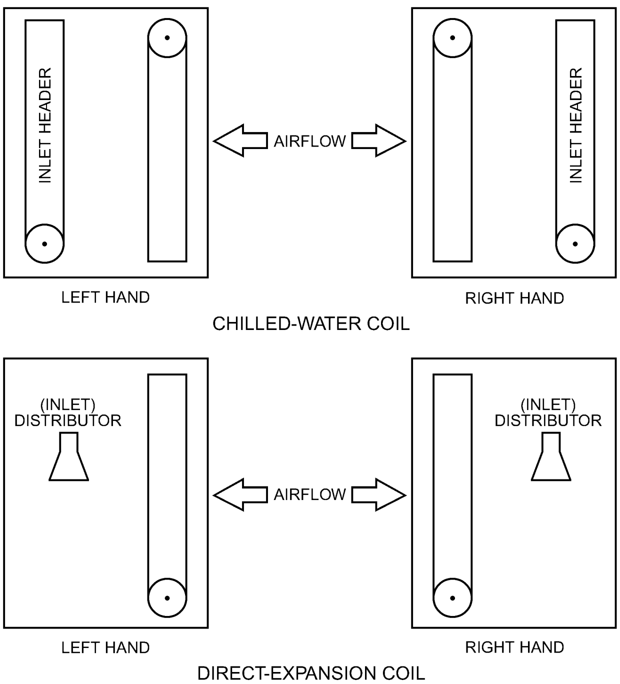

Fig. 3 Typical Coil Hand Designation

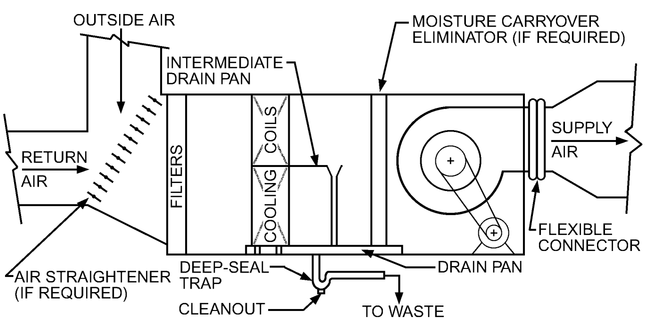

Fig. 4 Typical Arrangement of Cooling Coil Assembly in Built-Up or Packaged Central Station Air Handler

Factory-assembled central station air handlers incorporate most of the design features outlined for field built-up systems. These packaged units can generally accommodate various sizes, types, and row depths of cooling and heating coils to meet most job requirements. This usually eliminates the need for field built-up central systems, except on very large jobs.

The coil's design features (fin spacing, tube spacing, face height, type of fins), together with the amount of moisture on the coil and the degree of surface cleanliness, determine the air velocity at which condensed moisture blows off the coil. Generally, condensate water begins to be blown off a plate fin coil face at air velocities above 600 fpm. Water blowoff from coils into air ductwork external to the air-conditioning unit should be prevented. However, water blowoff is not usually a problem if coil fin heights are limited to 45 in. and the unit is set up to catch and dispose of condensate. When a number of coils are stacked one above another, condensate is carried into the airstream as it drips from one coil to the next. A downstream eliminator section could prevent this, but an intermediate drain pan and/or condensate trough (Figure 5) to collect the condensate and conduct it directly to the main drain pan is preferred. Extending downstream of the coil, each drain pan length should be at least one-half the coil height, and somewhat greater when coil airflow face velocities and/or humidity levels are higher.

When water is likely to carry over from the air-conditioning unit into external air ductwork, and no other means of prevention is provided, eliminator plates should be installed downstream of the coils. Usually, eliminator plates are not included in packaged units because other means of preventing carryover, such as space made available within the unit design for longer drain pan(s), are included in the design.

However, on sprayed-coil units, eliminators are usually included in the design. Such cooling and dehumidifying coils are sometimes sprayed with water to increase the rate of heat transfer, provide outlet air approaching saturation, and continually wash the surface of the coil. Coil sprays require a collecting tank, eliminators, and a recirculating pump (see Figure 6). Figure 6 also shows an air bypass, which helps a thermostat control maintain the humidity ratio by diverting a portion of the return air from the coil.

In field-assembled systems or factory-assembled central station air-handling units, fans are usually positioned downstream from the coil(s) in a draw-through arrangement. This arrangement provides acceptable airflow uniformity across the coil face more often than does the blow-through arrangement. In a blow-through arrangement, fan location upstream from the coils may require air baffles or

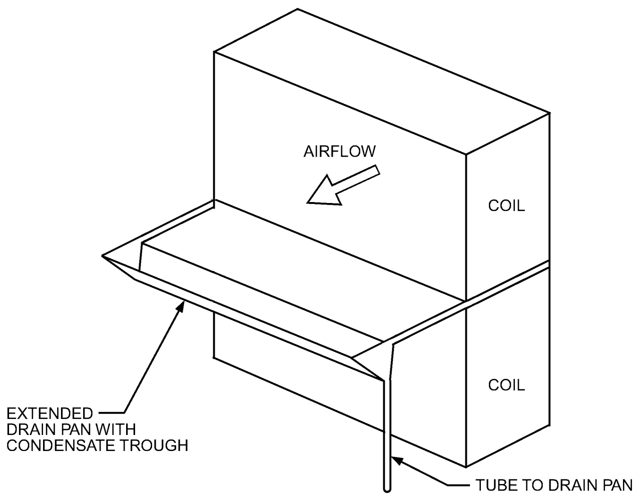

Fig. 5 Coil Bank Arrangement with Intermediate Condensate Pan

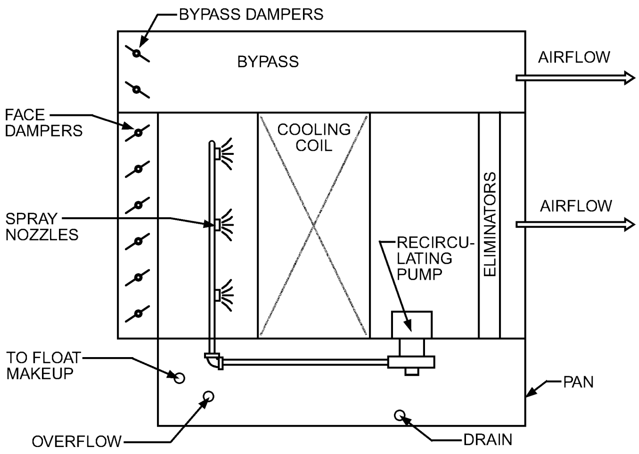

Fig. 6 Sprayed-Coil System with Air Bypass

diffuser plates between the fan discharge and the cooling coil to obtain uniform airflow. This is often the case in packaged multizone unit design. Airflow is considered to be uniform when measured flow across the entire coil face varies no more than 20%.

Air-cooling and dehumidifying coil frames, as well as all drain pans and troughs, should be of an acceptable corrosion-resistant material suitable for the system and its expected useful service life. The air handler's coil section enclosure should be corrosion-resistant; be properly double-wall insulated; and have adequate access doors for changing air filters, cleaning coils, adjusting flow control valves, and maintaining motors.

Where suction line risers are used for air-cooling coils in direct-expansion refrigeration systems, the suction line must be sized properly to ensure oil return from coil to compressor at minimum load conditions. Oil return is normally intrinsic with factory-assembled, self-contained air conditioners but must be considered for factory-assembled central station units or field-installed cooling coil banks where suction line risers are required and are assembled at the job site. Sizing, design, and arrangement of suction lines and their risers are described in Chapter 3 of the 2006 *ASHRAE Handbook—Refrigeration.*

COIL SELECTION

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

- Job requirements—cooling, dehumidifying, and the capacity required to properly balance with other system components (e.g., compressor equipment in the case of direct-expansion coils)
- Entering air dry-bulb and wet-bulb temperatures
- Available cooling media and operating temperatures
- Space and dimensional limitations
- Air and cooling fluid quantities, including distribution and limitations
- Allowable frictional resistances in air circuit (including coils)
- Allowable frictional resistances in cooling media piping system (including coils)
- Characteristics of individual coil designs and circuitry possibilities
- Individual installation requirements such as type of automatic control to be used; presence of corrosive atmosphere; design pressures; and durability of tube, fins, and frame material

Chapters 29 and 30 of the 2005 *ASHRAE Handbook—Fundamentals* contain information on load calculation.

Air quantity is affected by factors such as design parameters, codes, space, and equipment. Resistance through the air circuit influences fan power and speed. This resistance may be limited to allow the use of a given size fan motor, to keep operating expense low, or because of sound-level requirements. Air friction loss across the cooling coil (in summation with other series air-pressure drops for elements such as air filters, water sprays, heating coils, air grilles, and ductwork) determines the static pressure requirement for the complete airway system. The static pressure requirement is used in selecting fans and drives to obtain the design air quantity under operating conditions. See Chapter 18 for a description of fan selection.

The conditioned-air face velocity is determined by economic evaluation of initial and operating costs for the complete installation as influenced by (1) heat transfer performance of the specific coil surface type for various combinations of face areas and row depths as a function of air velocity; (2) air-side frictional resistance for the complete air circuit (including coils), which affects fan size, power, and sound-level requirements; and (3) condensate water carryover considerations. Allowable friction through the water or brine coil circuitry may be dictated by the head available from a given size pump and pump motor, as well as the same economic factors governing the air side made applicable to the water side. Additionally, the adverse effect of high cooling-water velocities on erosion-corrosion of tube walls is a major factor in sizing and circuitry to keep tube velocity below the recommended maximums. On larger coils, water pressure drop limits of 15 to 20 ft usually keep such velocities within acceptable limits of 2 to 4 fps, depending on circuit design.

Coil ratings are based on a uniform velocity. Design interference with uniform airflow through the coil makes predicting coil performance difficult as well as inaccurate. Such airflow interference may be caused by air entering at odd angles or by inadvertent blocking of a portion of the coil face. To obtain rated performance, the volumetric airflow quantity must be adjusted on the job to that at which the coil was rated and must be kept at that value. At start-up for air balance, the most common causes of incorrect airflow are the lack of altitude correction to standard air (where applicable) and ductwork problems. At commissioning, the most common causes of an air quantity deficiency are filter fouling and dirt or frost collection on the coils. These difficulties can be avoided through proper design, start-up checkout, and regular servicing.

The required total heat capacity of the cooling coil should be in balance with the capacity of other refrigerant system components such as the compressor, water chiller, condenser, and refrigerant liquid metering device. Chapter 43 of the 2006 *ASHRAE Handbook—Refrigeration* describes methods of estimating balanced system capacity under various operating conditions when using direct-expansion coils for both factory- and field-assembled systems.

For dehumidifying coils, it is important that the proper amount of surface area be installed to obtain the ratio of air-side sensible-to-total heat required to maintain air dry-bulb and wet-bulb temperatures in the conditioned space. This is an important consideration when preconditioning is done by reheat arrangement. The method for calculating the sensible and total heat loads and leaving air conditions at the coil to satisfy the sensible-to-total heat ratio required for the conditioned space is covered in Appendix D of *Cooling and Heating Load Calculation Principles* (Pedersen et al. 1998).

The same room air conditions can be maintained with different air quantities (including outside and return air) through a coil. However, for a given total air quantity with fixed percentages of outside and return air, there is only one set of air conditions leaving the coil that will precisely maintain room design air conditions. Once air quantity and leaving air conditions at the coil have been selected, there is usually only one combination of face area, row depth, and air face velocity for a given coil surface that will precisely maintain the required room ambient conditions. Therefore, in making final coil selections it is necessary to recheck the initial selection to ensure that the leaving air conditions, as calculated by a coil selection computer program or other procedure, will match those determined from the cooling load estimate.

Coil ratings and selections can be obtained from manufacturers' catalogs. Most catalogs contain extensive tables giving the performance of coils at various air and water velocities and entering humidity and temperatures. Most manufacturers provide computerized coil selection programs to potential customers. The final choice can then be made based on system performance and economic requirements.

Performance and Ratings

The long-term performance of an extended-surface air-cooling and dehumidifying coil depends on its correct design to specified conditions and material specifications, proper matching to other system components, proper installation, and proper maintenance as required.

In accordance with ARI *Standard* 410, Forced-Circulation Air-Cooling and Air-Heating Coils, dry-surface (sensible cooling) coils and dehumidifying coils (which both cool and dehumidify), particularly those for field-assembled coil banks or factory-assembled packaged units using different combinations of coils, are usually rated within the following parameters:

- Entering air dry-bulb temperature: 65 to 100°F
- Entering air wet-bulb temperature: 60 to 85°F (if air is not dehumidified in the application, select coils based on sensible heat transfer)
- Air face velocity: 200 to 800 fpm
- Evaporator refrigerant saturation temperature: 30 to 55°F at coil suction outlet (refrigerant vapor superheat at coil suction outlet is 6°F or higher)
- Entering chilled-water temperature: 35 to 65°F
- Water velocity: 1 to 8 fps
- For ethylene glycol solution: 1 to 6 fps, 0 to 90°F entering dry-bulb temperature, 60 to 80°F entering wet-bulb temperature, 10 to 60% aqueous glycol concentration by weight

The air-side ratio of sensible to total heat removed by dehumidifying coils varies in practice from about 0.6 to 1.0 (i.e., sensible heat is from 60 to 100% of the total, depending on the application). For information on calculating a dehumidifying coil's sensible heat ratio, see the section on Performance of Dehumidifying Coils, or Appendix D of *Cooling and Heating Load Calculation Principles* (Pedersen et al. 1998). For a given coil surface design and arrangement, the required sensible heat ratio may be satisfied by wide variations in and combinations of air face velocity, in-tube temperature, flow rate, entering air temperature, coil depth, and so forth, although the variations may be self-limiting. The maximum coil air face velocity should be limited to a value that prevents water carryover into the air ductwork. Dehumidifying coils for comfort applications are frequently selected in the range of 400 to 500 fpm air face velocity.

Operating ratings of dehumidifying coils for factory-assembled, self-contained air conditioners are generally determined in conjunction with laboratory testing for the system capacity of the complete unit assembly. For example, a standard rating point has been 33.4 cfm per 1000 Btu/h (or 400 cfm per ton of refrigeration effect), not to exceed 37.5 cfm per 100 Btu/h for unitary equipment. Refrigerant (e.g., R-22) duty would be 6 to 10°F superheat for an appropriate balance at 45°F saturated suction. For water coils, circuitry would operate at 4 fps, 42°F inlet water, 12°F rise (or 2 gpm per ton of refrigeration effect). The standard ratings at 80°F db and 67°F wb are representative of the entering air conditions encountered in many comfort operations. Although indoor conditions are usually lower than 67°F wb, it is usually assumed that introduction of outside air brings the air mixture to the cooling coil up to about 80°F db/67°F wb entering air design conditions.

Dehumidifying coils for field-assembled projects and central station air-handling units were formerly selected according to coil rating tables but are now selected by computerized selection programs. Either way, selecting coils from the load division indicated by the load calculation works satisfactorily for the usual human comfort applications. Additional design precautions and refinements are necessary for more exacting industrial applications and for all types of air conditioning in humid areas. One such refinement, the dual-path air process, uses a separate cooling coil to cool and dehumidify ventilation air before mixing it with recirculated air. This process dehumidifies what is usually the main source of moisture: makeup outside air. Condenser heat reclaim (when available) is another refinement required for some industrial applications and is finding greater use in commercial and comfort applications.

Airflow ratings are based on standard air of 0.075 lb/ft^3 at 70°F and a barometric pressure of 29.92 in. Hg. In some mountainous areas with a sufficiently large market, coil ratings and altitude-corrected psychrometrics are available for their particular altitudes.

When checking the operation of dehumidifying coils, climatic conditions must be considered. Most problems are encountered at light-load conditions, when the cooling requirement is considerably less than at design conditions. In hot, dry climates, where the outside dew point is consistently low, dehumidifying is not generally a problem, and the light-load design point condition does not pose any special problems. In hot, humid climates, the light-load condition has a higher proportion of moisture and a correspondingly lower proportion of sensible heat. The result is higher dew points in the conditioned spaces during light-load conditions unless a special means for controlling inside dew points (e.g., reheat or dual path) is used.

Fin surface freezing at light loads should be avoided. Freezing occurs when a dehumidification coil's surface temperature falls below 32°F. Freezing does not occur with standard coils for comfort installations unless the refrigerant evaporating temperature at the coil outlet is below 25 to 28°F saturated; the exact value depends on the design of the coil, its operating dew point, and the amount of loading. With coil and condensing units to balance at low temperatures at peak loads (not a customary design choice), freezing may occur when load suddenly decreases. The possibility of this type of surface freezing is greater if a bypass is used because it causes less air to be passed through the coil at light loads.

AIRFLOW RESISTANCE

A cooling coil's airflow resistance (air friction) depends on the tube pattern and fin geometry (tube size and spacing, fin configuration, and number of in-line or staggered rows), coil face velocity, and amount of moisture on the coil. The coil air friction may also be affected by the degree of aerodynamic cleanliness of the coil core; burrs on fin edges may increase coil friction and increase the tendency to pocket dirt or lint on the faces. A completely dry coil, removing only sensible heat, offers approximately one-third less resistance to airflow than a dehumidifying coil removing both sensible and latent heat.

For a given surface and airflow, increasing the number of rows or fins increases airflow resistance. Therefore, final selection involves economic balancing of the initial cost of the coil against the operating costs of the coil geometry combinations available to adequately meet the performance requirements.

The aluminum fin surfaces of new dehumidifying coils tend to inhibit condensate sheeting action until they have aged for a year. Hydrophilic aluminum fin surface coatings reduce water droplet surface tension, producing a more evenly dispersed wetted surface action at initial start-up. Manufacturers have tried different methods of applying such coatings, including dipping the coil into a tank, coating the fin stock material, or subjecting the material to a chemical etching process. Tests have shown as much as a 30% reduction in air pressure drop across a hydrophilic coil as opposed a new untreated coil.

HEAT TRANSFER

The heat transmission rate of air passing over a clean tube (with or without extended surface) to a fluid flowing within it is impeded principally by three thermal resistances: (1) surface air-side film thermal resistance from the air to the surface of the exterior fin and tube assembly; (2) metal thermal resistance to heat conductance through the exterior fin and tube assembly; and (3) in-tube fluid-side film thermal resistance, which impedes heat flow between the internal surface of the metal and the fluid flowing within the tube. For some applications, an additional thermal resistance is factored in to account for external and/or internal surface fouling. Usually, the combination of metal and tube-side film resistance is considerably lower than the air-side surface resistance.

For a reduction in thermal resistance, the fin surface is fabricated with die-formed corrugations instead of the traditional flat design. At low airflows or wide fin spacing, the air-side transfer coefficient is virtually the same for flat and corrugated fins. Under normal comfort conditioning operation, the corrugated fin surface is designed to reduce the boundary air film thickness by undulating the passing airstream within the coil; this produces a marked improvement in heat transfer without much airflow penalty. Further fin enhancements, including louvered and lanced fin designs, have been driven by the desire to duplicate throughout the coil depth the thin boundary air film characteristic of the fin's leading edge. Louvered fin design maximizes the number of fin surface leading edges throughout the entire secondary surface area and increases the external secondary surface area A_s through the multiplicity of edges.

Where an application allows economical use of coil construction materials, the mass and size of the coil can be reduced when boundary air and water films are lessened. For example, the exterior surface resistance can be reduced to nearly the same as the fluid-side resistance by using lanced and/or louvered fins. External as well as internal tube fins (or internal turbulators) can economically decrease overall heat transfer surface resistances. Also, water sprays applied to a flat fin coil surface may increase overall heat transfer slightly, although they may better serve other purposes such as air and coil cleaning.

Heat transfer between the cooling medium and the airstream across a coil is influenced by the following variables:

- Temperature difference between fluids
- Design and surface arrangement of the coil
- Velocity and character of the airstream
- Velocity and character of the in-tube coolant

With water coils, only the water temperature rises. With coils of volatile refrigerants, an appreciable pressure drop and a corresponding change in evaporating temperature through the refrigerant circuit often occur. Alternative refrigerants to R-22, such as R-407C, which has a temperature glide, will have an evaporation temperature rise of 7 to 12°F through the evaporator. This must be considered in design and performance calculation of the coil. A compensating pressure drop in the coil may partially, or even totally, compensate for the low-side temperature glide of a zeotropic refrigerant blend. Rating direct-expansion coils is further complicated by the refrigerant evaporating in part of the circuit and superheating in the remainder. Thus, for halocarbon refrigerants, a cooling coil is tested and rated with a specific distributing and liquid-metering device, and the capacities are stated with the superheat condition of the leaving vapor.

At a given air mass velocity, performance depends on the turbulence of airflow into the coil and the uniformity of air distribution over the coil face. The latter is necessary to obtain reliable test ratings and realize rated performance in actual installations. Air resistance through the coils helps distribute air properly, but the effect is frequently inadequate where inlet duct connections are brought in at sharp angles to the coil face. Reverse air currents may pass through a portion of the coils. These currents reduce capacity but can be avoided with proper inlet air vanes or baffles. Air blades may also be required. Remember that coil performance ratings (ARI *Standard* 410) represent optimum conditions resulting from adequate and reliable laboratory tests (ASHRAE *Standard* 33).

For cases when available data must be extended, for arriving at general design criteria for a single, unique installation, or for understanding the calculation progression, the following material and illustrative examples for calculating cooling coil performance are useful guides.

PERFORMANCE OF SENSIBLE COOLING COILS

The performance of sensible cooling coils depends on the following factors. See the section on Symbols for an explanation of the variables.

- The overall coefficient U_o of sensible heat transfer between airstream and coolant fluid
- The mean temperature difference Δt_m between airstream and coolant fluid
- The physical dimensions of and data for the coil (such as coil face area A_a and total external surface area A_o) with characteristics of the heat transfer surface

The sensible heat cooling capacity q_{td} of a given coil is expressed by the following equation:

$$q_{td} = U_o F_s A_a N_r \Delta t_m \tag{1a}$$

with

$$F_s = A_o / A_a N_r \tag{1b}$$

Assuming no extraneous heat losses, the same amount of sensible heat is lost from the airstream:

$$q_{td} = w_a c_p (t_{a1} - t_{a2}) \tag{2a}$$

with

$$w_a = \rho_a A_a V_a \tag{2b}$$

The same amount of sensible heat is absorbed by the coolant; for a nonvolatile type, it is

$$q_{td} = w_r c_r (t_{r2} - t_{r1}) \tag{3}$$

For a nonvolatile coolant in thermal counterflow with the air, the mean temperature difference in Equation (1a) is expressed as

$$\Delta t_m = \frac{(t_{a1} - t_{r2}) - (t_{a2} - t_{r1})}{\ln[(t_{a1} - t_{r2})/(t_{a2} - t_{r1})]} \tag{4}$$

Proper temperature differences for various crossflow situations are given in many texts, including Mueller (1973). These calculations are based on various assumptions, among them that U for the total external surface is constant. Although this assumption is generally not valid for multirow coils, using crossflow temperature differences from Mueller (1973) or other texts should be preferable to Equation (4), which applies only to counterflow. However, using the log mean temperature difference is widespread.

The overall heat transfer coefficient U_o for a given coil design, whether bare-pipe or finned-type, with clean, nonfouled surfaces, consists of the combined effect of three individual heat transfer coefficients:

- The **film coefficient** f_a of sensible heat transfer between air and the external surface of the coil
- The **unit conductance** $1/R_{md}$ of the coil material (i.e., tube wall, fins, tube-to-fin thermal resistance)
- The **film coefficient** f_r of heat transfer between the internal coil surface and the coolant fluid within the coil

These three individual coefficients acting in series form an overall coefficient of heat transfer in accordance with the material given in Chapters 3 and 23 of the 2005 *ASHRAE Handbook—Fundamentals*.

For a bare-pipe coil, the overall coefficient of heat transfer for sensible cooling (without dehumidification) can be expressed by a simplified basic equation:

$$U_o = \frac{1}{(1/f_a) + (D_o - D_i)/24k + (B/f_r)} \tag{5a}$$

When pipe or tube walls are thin and of high-conductivity material (as in typical heating and cooling coils), the term $(D_o - D_i)/24k$ in Equation (5a) frequently becomes negligible and is generally disregarded. (This effect in typical bare-pipe cooling coils seldom exceeds 1 to 2% of the overall coefficient.) Thus, the overall coefficient for bare pipe in its simplest form is

$$U_o = \frac{1}{(1/f_a) + (B/f_r)} \tag{5b}$$

For finned coils, the equation for the overall coefficient of heat transfer can be written

$$U_o = \frac{1}{(1/\eta f_a) + (B/f_r)} \tag{5c}$$

where the **fin effectiveness** η allows for the resistance to heat flow encountered in the fins. It is defined as

$$\eta = (EA_s + A_p)/A_o \tag{6}$$

For typical cooling surface designs, the surface ratio B ranges from about 1.03 to 1.15 for bare-pipe coils and from 10 to 30 for finned coils. Chapter 3 of the 2005 *ASHRAE Handbook—Fundamentals* describes how to estimate fin efficiency and calculate the tube-side heat transfer coefficient f_r for nonvolatile fluids. Table 2 in ARI *Standard* 410 lists thermal conductivity k of standard coil materials.

Estimating the air-side heat transfer coefficient f_a is more difficult because well-verified general predictive techniques are not available. Hence, direct use of experimental data is usually necessary. For plate fin coils, some correlations that satisfy several data sets are available (Kusuda 1970; McQuiston 1981). Webb (1980) reviewed air-side heat transfer and pressure drop correlations for various geometries. Mueller (1973) and Chapter 3 of the 2005 *ASHRAE Handbook—Fundamentals* provide guidance on this subject.

For analyzing a given heat exchanger, the concept of **effectiveness** is useful. Expressions for effectiveness have been derived for various flow configurations and can be found in Kusuda (1970) and Mueller (1973). The cooling coils covered in this chapter actually involve various forms of crossflow. However, the case of counterflow is addressed here to illustrate the value of this concept. The air-side effectiveness E_a for counterflow heat exchangers is given by the following equations:

$$q_{td} = w_a c_p (t_{a1} - t_{r1}) E_a \tag{7a}$$

with

$$E_a = \frac{t_{a1} - t_{a2}}{t_{a1} - t_{r1}} \tag{7b}$$

or

$$E_a = \frac{1 - e^{-c_o(1-M)}}{1 - Me^{-c_o(1-M)}} \tag{7c}$$

with

$$c_o = \frac{A_o U_o}{w_a c_p} = \frac{F_s N_r U_o}{60 \rho_a V_a c_p} \tag{7d}$$

and

$$M = \frac{w_a c_p}{w_r c_r} = \frac{60 \rho_a A_a V_a c_p}{w_r c_r} \tag{7e}$$

Note the following two special conditions:

If $M = 0$, then $E_a = 1 - e^{-c_o}$
If $M \geq 1$, then

$$E_a = \frac{1}{(1/c_o) + 1}$$

With a given design and arrangement of heat transfer surface used as cooling coil core material for which basic physical and heat transfer data are available to determine U_o from Equations (5a), (5b), and (5c), the selection, sizing, and performance calculation of sensible cooling coils for a particular application generally fall into either of two categories:

1. Heat transfer surface area A_o or coil row depth N_r for a specific coil size is required and initially unknown. Sensible cooling capacity q_{td}, flow rates for both air and coolant, entrance and exit temperatures of both fluids, and mean temperature difference between fluids are initially known or can be assumed or determined from Equations (2a), (3), and (4). A_o or N_r can then be calculated directly from Equation (1a).
2. Sensible cooling capacity q_{td} for a specific coil is required and initially unknown. Face area and heat transfer surface area are known or can be readily determined. Flow rates and entering temperatures of air and coolant are also known. Mean temperature difference Δt_m is unknown, but its determination is unnecessary to calculate q_{td}, which can be found directly by solving Equation (7a). Equation (7a) also provides a basic means of determining q_{td} for a given coil or related family of coils over the complete rating ranges of air and coolant flow rates and operating temperatures.

The two categories of application problems are illustrated in Examples 1 and 2, respectively:

Example 1. Standard air flowing at a mass rate equivalent to 9000 cfm is to be cooled from 85 to 75°F, using 330 lb/min chilled water supplied at 50°F in thermal counterflow arrangement. Assuming an air face velocity of V_a = 600 fpm and no air dehumidification, calculate coil face area A_a, sensible cooling capacity q_{td}, required heat transfer surface area A_o, coil row depth N_r, and coil air-side pressure drop Δp_{st} for a clean, nonfouled, thin-walled bare copper tube surface design for which the following physical and performance data have been predetermined:

B = surface ratio = 1.07
c_p = 0.24 Btu/lb·°F
c_r = 1.0 Btu/lb·°F
F_s = (external surface area)/(face area)(rows deep) = 1.34
f_a = 15 Btu/h·ft²·°F
f_r = 800 Btu/h·ft²·°F
$\Delta p_{st}/N_r$ = 0.027 in. of water/number of coil rows
ρ_a = 0.075 lb/ft³

Solution: Calculate the coil face area required.

$$A_a = 9000/600 = 15 \text{ ft}^2$$

Neglecting the effect of tube wall, from Equation (5b),

$$U_o = \frac{1}{(1/15) + (1.07/800)} = 14.7 \text{ Btu/h}\cdot\text{ft}^2\cdot{}^\circ\text{F}$$

From Equations (2a) and (2b), the sensible cooling capacity is

$$q_{td} = 60 \times 0.075 \times 15 \times 600 \times 0.24(85 - 75) = 97{,}200 \text{ Btu/h}$$

From Equation (3),

$$t_{r2} = 50 + 97{,}200/(330 \times 60 \times 1)(1000 \times 2.5 \times 4.18) = 54.9°\text{F}$$

From Equation (4),

$$\Delta t_m = \frac{(85 - 54.9) - (75 - 50)}{\ln[(85 - 54.9)/(75 - 50)]} = 27.5°\text{F}$$

From Equations (1a) and (1b), the surface area required is

$$A_o = 97{,}200/(14.7 \times 27.5) = 240 \text{ ft}^2 \text{ external surface}$$

From Equation (1b), the required row depth is

$$N_r = 240/(1.34 \times 15) = 11.9 \text{ rows deep}$$

The installed 15 ft² coil face, 12 rows deep, slightly exceeds the required capacity. The air-side pressure drop for the installed row depth is then

$$\Delta p_{st} = (\Delta p_{st}/N_r)N_r = 0.027 \times 12 = 0.32 \text{ in. of water at 70°F}$$

In this example, for some applications where such items as V_a, w_r, t_{r1}, and f_r may be arbitrarily varied with a fixed design and arrangement of heat transfer surface, a trade-off between coil face area A_a and coil row depth N_r is sometimes made to obtain alternative coil selections that produce the same sensible cooling capacity q_{td}. For example, an eight-row coil could be selected, but it would require a larger face area A_a with lower air face velocity V_a and a lower air-side pressure drop Δp_{st}.

Example 2. An air-cooling coil using a finned tube-type heat transfer surface has physical data as follows:

$$\begin{aligned} A_a &= 10 \text{ ft}^2 \\ A_o &= 800 \text{ ft}^2 \text{ external} \\ B &= \text{surface ratio} = 20 \\ F_s &= \text{(external surface area)/(face area)(rows deep)} = 27 \\ N_r &= 3 \text{ rows deep} \end{aligned}$$

Air at a face velocity of V_a = 800 fpm and 95°F entering air temperature is to be cooled by 15 gpm of well water supplied at 55°F. Calculate the sensible cooling capacity q_{td}, leaving air temperature t_{a2}, leaving water temperature t_{r2}, and air-side pressure drop Δp_{st}. Assume clean and nonfouled surfaces, thermal counterflow between air and water, no air dehumidification, standard barometric air pressure, and that the following data are available or can be predetermined:

$$\begin{aligned} c_p &= 0.24 \text{ Btu/lb·°F} \\ c_r &= 1.0 \text{ Btu/lb·°F} \\ f_a &= 17 \text{ Btu/h·ft}^2\text{·°F} \\ f_r &= 500 \text{ Btu/h·ft}^2\text{·°F} \\ \eta &= \text{fin effectiveness} = 0.9 \\ \Delta p_{st}/N_r &= 0.22 \text{ in. of water/number of coil rows} \\ \rho_a &= 0.075 \text{ lb/ft}^3 \\ \rho_w &= 62.4 \text{ lb/ft}^3 \text{ (8.34 lb/gal)} \end{aligned}$$

Solution: From Equation (5c),

$$U_o = \frac{1}{1/(0.9 \times 17) + (20/500)} = 9.5 \text{ Btu/h·ft}^2\text{·°F}$$

From Equations (7d) and (2b),

$$c_o = \frac{800 \times 9.5}{60 \times 0.075 \times 10 \times 800 \times 0.24} = 0.88$$

From Equation (7e),

$$M = \frac{60 \times 0.075 \times 10 \times 800 \times 0.24}{15 \times 60 \times 8.34 \times 1} = 1.15$$

Substituting in,

$$-c_o(1 - M) = -0.88(1 - 1.15) = 0.132$$

From Equation (7c),

$$E_a = \frac{1 - e^{0.132}}{1 - 1.15e^{0.132}} = 0.452$$

From Equation (7a), the sensible cooling capacity is

$$q_{td} = 60 \times 0.075 \times 10 \times 800 \times 0.24(95 - 55) \times 0.452 = 156{,}000 \text{ Btu/h}$$

From Equation (2a), the leaving air temperature is

$$t_{a2} = 95 - \frac{156{,}000}{60 \times 0.075 \times 10 \times 800 \times 0.24} = 76.9°\text{F}$$

From Equation (3), the leaving water temperature is

$$t_{r2} = 55 + \frac{156{,}000}{15 \times 60 \times 8.34 \times 1} = 75.8°\text{F}$$

The air-side pressure drop is

$$\Delta p_{st} = 0.22 \times 3 = 0.66 \text{ in. of water}$$

The preceding equations and examples demonstrate the method for calculating thermal performance of sensible cooling coils that operate with a dry surface. However, when cooling coils operate wet or act as dehumidifying coils, performance cannot be predicted without including the effect of air-side moisture (latent heat) removal.

PERFORMANCE OF DEHUMIDIFYING COILS

A dehumidifying coil normally removes both moisture and sensible heat from entering air. In most air-conditioning processes, the air to be cooled is a mixture of water vapor and dry air gases. Both lose sensible heat when in contact with a surface cooler than the air. Latent heat is removed through condensation only on the parts of the coil where the surface temperature is lower than the dew point of the air passing over it. Figure 2 in Chapter 4 shows the assumed psychrometric conditions of this process. As the leaving dry-bulb temperature drops below the entering dew-point temperature, the difference between leaving dry-bulb temperature and leaving dew point for a given coil, airflow, and entering air condition is lessened.

When the coil starts to remove moisture, the cooling surfaces carry both the sensible and latent heat load. As the air approaches saturation, each degree of sensible cooling is nearly matched by a corresponding degree of dew-point decrease. The latent heat removal per degree of dew-point change is considerably greater. The following table compares the amount of moisture removed from air at standard barometric pressure that is cooled from 60 to 59°F at both wet and dry conditions.

Dew Point	h_s, Btu/lb	Dry Bulb	h_a, Btu/lb
60°F	26.467	60°F	14.415
59°F	25.792	59°F	14.174
Difference	0.675	Difference	0.241

Note: These numerical values conform to Table 2 in Chapter 6 of the 2005 *ASHRAE Handbook—Fundamentals*.

For volatile refrigerant coils, the refrigerant distributor assembly must be tested at the higher and lower capacities of its rated range. Testing at lower capacities checks whether the refrigerant distributor provides equal distribution and whether the control is able to modulate without hunting. Testing at higher capacities checks the maximum feeding capacity of the flow control device at the greater pressure drop that occurs in the coil system.

Most manufacturers develop and produce their own performance rating tables using data obtained from suitable tests. ASHRAE *Standard* 33 specifies the acceptable method of lab-testing coils. ARI *Standard* 410 gives a method for rating thermal performance of dehumidifying coils by extending data from laboratory tests on prototypes to other operating conditions, coil sizes, and row depths. To account for simultaneous transfer of both sensible and latent heat from the airstream to the surface, ARI *Standard* 410 uses essentially the same method for arriving at cooling and dehumidifying coil thermal performance as determined by McElgin and Wiley (1940) and described in the context of *Standard* 410 by Anderson (1970). In systems operating at partial flow such as in thermal storage, accurate performance predictions for 2300 < Re < 4000 flows have been obtained by using the Gnielinski correlation. This work was presented in detail comparable to *Standard* 410 by Mirth et al. (1993).

The potential or driving force for transferring total heat q_t from the airstream to the tube-side coolant is composed of two components in series heat flow: (1) an air-to-surface air enthalpy difference $(h_a - h_s)$ and (2) a surface-to-coolant temperature difference $(t_s - t_r)$.

Figure 7 is a typical thermal diagram for a coil in which the air and a nonvolatile coolant are arranged in counterflow. The top and bottom lines in the diagram indicate, respectively, changes across the coil in the airstream enthalpy h_a and the coolant temperature t_r. To illustrate continuity, the single middle line in Figure 7 represents both surface temperature t_s and the corresponding saturated air enthalpy h_s, although the temperature and air enthalpy scales do not actually coincide as shown. The differential surface area dA_w represents any specific location within the coil thermal diagram where operating conditions are such that the air-surface interface temperature t_s is lower than the local air dew-point temperature. Under these conditions, both sensible and latent heat are removed from the airstream, and the cooler surface actively condenses water vapor.

Neglecting the enthalpy of condensed water vapor leaving the surface and any radiation and convection losses, the total heat lost from the airstream in flowing over dA_w is

$$dq_t = -w_a(dh_a) \tag{8}$$

This same total heat is transferred from the airstream to the surface interface. According to McElgin and Wiley (1940),

$$dq_t = \frac{(h_a - h_s)dA_w}{c_p R_{aw}} \tag{9}$$

The total heat transferred from the air-surface interface across the surface elements and into the coolant is equal to that given in Equations (8) and (9):

$$dq_t = \frac{(t_s - t_r)dA_w}{R_{mw} + R_r} \tag{10}$$

The same quantity of total heat is also gained by the nonvolatile coolant in passing across dA_w:

$$dq_t = -w_r c_r(dt_r) \tag{11}$$

If Equations (9) and (10) are equated and the terms rearranged, an expression for the coil characteristic C is obtained:

$$C = \frac{R_{mw} + R_r}{c_p R_{aw}} = \frac{t_s - t_r}{h_a - h_s} \tag{12}$$

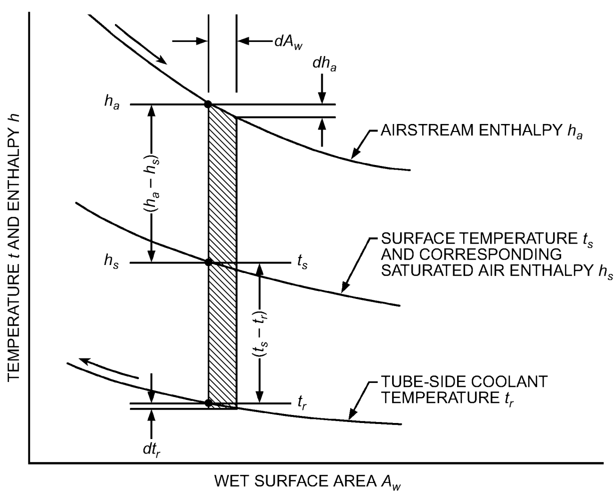

Fig. 7 Two-Component Driving Force Between Dehumidifying Air and Coolant

Equation (12) shows the basic relationship of the two components of the driving force between air and coolant in terms of three principal thermal resistances. For a given coil, these three resistances of air, metal, and in-tube fluid (R_{aw}, R_{mw}, and R_r) are usually known or can be determined for the particular application, which gives a fixed value for C. Equation (12) can then be used to determine point conditions for the interrelated values of airstream enthalpy h_a, coolant temperature t_r, surface temperature t_s, and enthalpy h_s of saturated air corresponding to the surface temperature. When both t_s and h_s are unknown, a trial-and-error solution is necessary; however, this can be solved graphically by a surface temperature chart such as Figure 8.

Figure 9 shows a typical thermal diagram for a portion of the coil surface when it is operating dry. The illustration is for counterflow with a halocarbon refrigerant. The diagram at the top of the figure illustrates a typical coil installation in an air duct with tube passes circuited countercurrent to airflow. Locations of the entering and leaving boundary conditions for both air and coolant are shown.

The thermal diagram in Figure 9 is the same type as in Figure 7, showing three lines to illustrate local conditions for the air, surface, and coolant throughout a coil. The dry-wet boundary conditions are located where the coil surface temperature t_{sb} equals the entering air dew-point temperature t''_{a1}. Thus, the surface area A_d to the left of this boundary is dry, with the remainder A_w of the coil surface area operating wet.

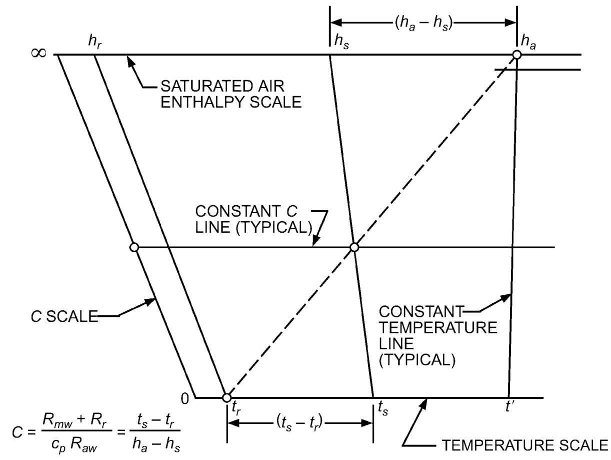

Fig. 8 Surface Temperature Chart

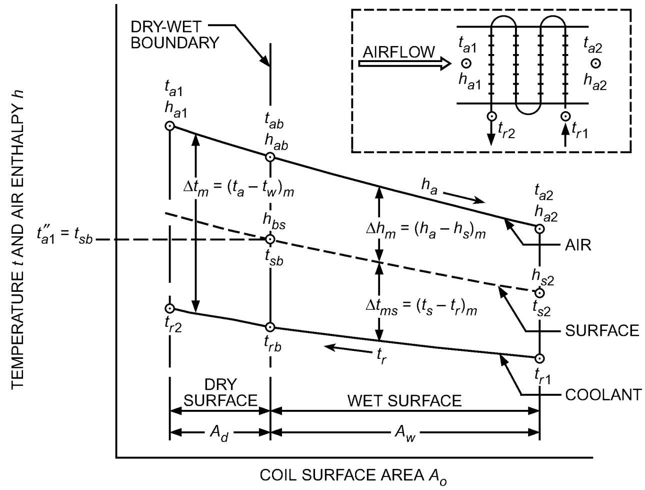

Fig. 9 Thermal Diagram for General Case When Coil Surface Operates Partially Dry

When using fluids or halocarbon refrigerants in a thermal counterflow arrangement as illustrated in Figure 9, the dry-wet boundary conditions can be determined from the following relationships:

$$y = \frac{t_{r2} - t_{r1}}{h_{a1} - h_{a2}} = \frac{w_a}{w_r c_r} \tag{13}$$

$$h_{ab} = \frac{t''_{a1} - t_{r2} + y h_{a1} + C h''_{a1}}{C + y} \tag{14}$$

The value of h_{ab} from Equation (14) serves as an index of whether the coil surface is operating fully wetted, partially dry, or completely dry, according to the following three limits:

1. If $h_{ab} \geq h_{a1}$, the surface is fully wetted.
2. If $h_{a1} > h_{ab} > h_{a2}$, the surface is partially dry.
3. If $h_{ab} \leq h_{a2}$, the surface is completely dry.

Other dry-wet boundary properties are then determined:

$$t_{sb} = t''_{a1} \tag{15}$$

$$t_{ab} = t_{a1} - (h_{a1} - h_{ab})/c_p \tag{16}$$

$$t_{rb} = t_{r2} - y c_p (t_{a1} - t_{ab}) \tag{17}$$

The dry surface area A_d required and capacity q_{td} are calculated by conventional sensible heat transfer relationships, as follows.

The overall thermal resistance R_o comprises three basic elements:

$$R_o = R_{ad} + R_{md} + R_r \tag{18}$$

with

$$R_r = B/f_r \tag{19}$$

The mean difference between air dry-bulb temperature and coolant temperature, using symbols from Figure 9, is

$$\Delta t_m = \frac{(t_{a1} - t_{r2}) - (t_{ab} - t_{rb})}{\ln[(t_{a1} - t_{r2})/(t_{ab} - t_{rb})]} \tag{20}$$

The dry surface area required is

$$A_d = \frac{q_{td} R_o}{\Delta t_m} \tag{21}$$

The air-side total heat capacity is

$$q_{td} = w_a c_p (t_{a1} - t_{ab}) \tag{22a}$$

From the coolant side,

$$q_{td} = w_r c_r (t_{r2} - t_{rb}) \tag{22b}$$

The wet surface area A_w and capacity q_{tw} are determined by the following relationships, using terminology in Figure 9.

For a given coil size, design, and arrangement, the fixed value of the coil characteristic C can be determined from the ratio of the three prime thermal resistances for the job conditions:

$$C = \frac{R_{mw} + R_r}{c_p R_{aw}} \tag{23}$$

Knowing coil characteristic C for point conditions, the interrelations between airstream enthalpy h_a, coolant temperature t_r, and surface temperature t_s and its corresponding enthalpy of saturated air h_s can be determined by using a surface temperature chart (Figure 8) or by a trial-and-error procedure using Equation (24):

$$C = \frac{t_{sb} - t_{rb}}{h_{ab} - h_{sb}} = \frac{t_{s2} - t_{r1}}{h_{a2} - h_{s2}} \tag{24}$$

The mean effective difference in air enthalpy between airstream and surface from Figure 9 is

$$\Delta h_m = \frac{(h_{ab} - h_{sb}) - (h_{a2} - h_{s2})}{\ln[(h_{ab} - h_{sb})/(h_{a2} - h_{s2})]} \tag{25}$$

Similarly, the mean temperature difference between surface and coolant is

$$\Delta t_{ms} = \frac{(t_{sb} - t_{rb}) - (t_{s2} - t_{r1})}{\ln[(t_{sb} - t_{rb})/(t_{s2} - t_{r1})]} \tag{26}$$

The wet surface area required, calculated from air-side enthalpy difference, is

$$A_w = \frac{q_{tw} R_{aw} c_p}{\Delta h_m} \tag{27a}$$

Calculated from the coolant-side temperature difference,

$$A_w = \frac{q_{tw}(R_{mw} + R_r)}{\Delta t_{ms}} \tag{27b}$$

The air-side total heat capacity is

$$q_{tw} = w_a[h_{a1} - (h_{a2} + h_{fw})] \tag{28a}$$

The enthalpy h_{fw} of condensate removed is

$$h_{fw} = (W_1 - W_2) c_{pw} (t'_{a2} - 32) \tag{28b}$$

where c_{pw} = specific heat of water = 1.0 Btu/lb$_w$·°F.

Note that h_{fw} for normal air-conditioning applications is about 0.5% of the airstream enthalpy difference $(h_{a1} - h_{a2})$ and is usually neglected.

The coolant-side heat capacity is

$$q_{tw} = w_r c_r (t_{rb} - t_{r1}) \tag{28c}$$

The total surface area requirement of the coil is

$$A_o = A_d + A_w \tag{29}$$

The total heat capacity for the coil is

$$q_t = q_{td} + q_{tw} \tag{30}$$

The leaving air dry-bulb temperature is found by the method illustrated in Figure 10, which represents part of a psychrometric chart showing the air saturation curve and lines of constant air enthalpy closely corresponding to constant wet-bulb temperature lines.

For a given coil and air quantity, a straight line projected through the entering and leaving air conditions intersects the air saturation curve at a point denoted as the effective coil surface temperature $t_{\bar{s}}$. Thus, for fixed entering air conditions t_{a1} and h_{a1} and a given effective surface temperature t_s, leaving air dry bulb t_{a2} increases but is still located on this straight line if air quantity is increased or coil depth is reduced. Conversely, a decrease in air quantity or an increase in coil depth produces a lower t_{a2} that is still located on the same straight-line segment.

An index of the air-side effectiveness is the heat transfer exponent c, defined as

$$c = \frac{A_o}{w_a c_p R_{ad}} \tag{31}$$

This exponent c, sometimes called the number of air-side transfer units NTU$_a$, is also defined as

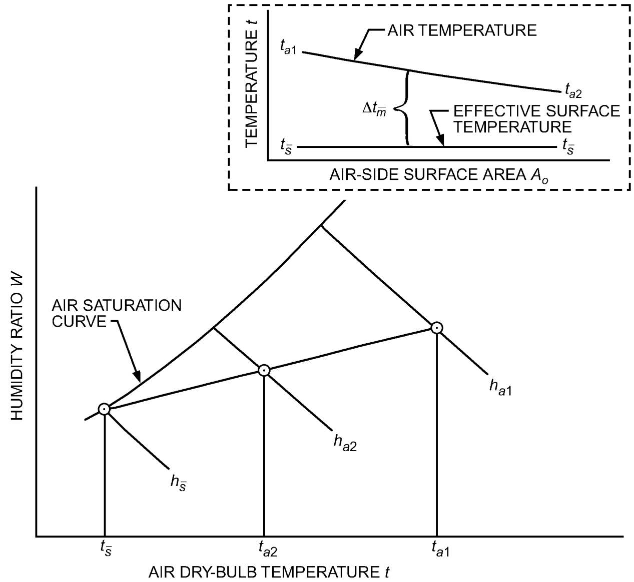

Fig. 10 Leaving Air Dry-Bulb Temperature Determination for Air-Cooling and Dehumidifying Coils

$$c = \frac{t_{a1} - t_{a2}}{\Delta t_{\bar{m}}} \tag{32}$$

The temperature drop $(t_{a1} - t_{a2})$ of the airstream and mean temperature difference $\Delta t_{\bar{m}}$ between air and effective surface in Equation (32) are illustrated at the top of Figure 10.

Knowing the exponent c and entering and leaving enthalpies h_{a1} and h_{a2} for the airstream, the enthalpy of saturated air $h_{\bar{s}}$ corresponding to effective surface temperature $t_{\bar{s}}$ is calculated as follows:

$$h_{\bar{s}} = h_{a1} - \frac{h_{a1} - h_{a2}}{1 - e^{-c}} \tag{33}$$

After finding the value of $t_{\bar{s}}$ that corresponds to $h_{\bar{s}}$ from the saturated air enthalpy tables, the leaving air dry-bulb temperature can be determined:

$$t_{a2} = t_{\bar{s}} + e^{-c}(t_{a1} - t_{\bar{s}}) \tag{34}$$

The air-side sensible heat ratio SHR can then be calculated:

$$\text{SHR} = \frac{c_p(t_{a1} - t_{a2})}{h_{a1} - h_{a2}} \tag{35}$$

For thermal performance of a coil to be determined from the foregoing relationships, values of the following three principal resistances to heat flow between air and coolant must be known:

- Total metal thermal resistances across the fin R_f and tube assembly R_t for both dry R_{md} and wet R_{mw} surface operation
- Air-film thermal resistances R_{ad} and R_{aw} for dry and wet surfaces, respectively
- Tube-side coolant film thermal resistance R_r

In ARI *Standard* 410, metal thermal resistance R_m is calculated based on the physical data, material, and arrangement of the fin and tube elements, together with the fin efficiency E for the specific fin configuration. R_m is variable as a weak function of the effective air-side heat transfer coefficient f_a for a specific coil geometry, as illustrated in Figure 11.

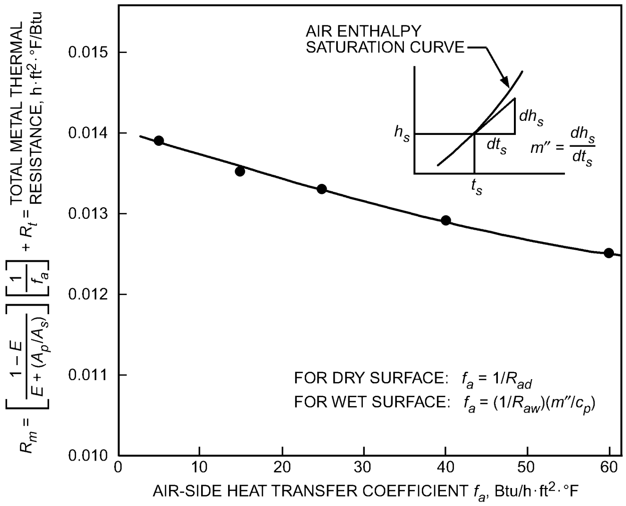

Fig. 11 Typical Total Metal Thermal Resistance of Fin and Tube Assembly

For wetted-surface application, Brown (1954), with certain simplifying assumptions, showed that f_a is directly proportional to the rate of change m'' of saturated air enthalpy h_s with the corresponding surface temperature t_s. This slope m'' of the air enthalpy saturation curve is illustrated in the small inset graph at the top of Figure 11.

The abscissa for f_a in the main graph of Figure 11 is an effective value, which, for a dry surface, is the simple thermal resistance reciprocal $1/R_{ad}$. For a wet surface, f_a is the product of the thermal resistance reciprocal $1/R_{aw}$ and the multiplying factor m''/c_p. ARI *Standard* 410 outlines a method for obtaining a mean value of m''/c_p for a given coil and job condition. The total metal resistance R_m in Figure 11 includes the resistance R_t across the tube wall. For most coil designs, R_t is quite small compared to the resistance R_f through the fin metal.

The air-side thermal resistances R_{ad} and R_{aw} for dry and wet surfaces, together with their respective air-side pressure drops $\Delta p_{st}/N_r$ and $\Delta p_{sw}/N_r$, are determined from tests on a representative coil model over the full range in the rated airflow. Typical plots of experimental data for these four performance variables versus coil air face velocity V_a at 70°F are illustrated in Figure 12.

If water is used as the tube-side coolant, the heat transfer coefficient f_r is calculated from Equation (8) in ARI *Standard* 410. For evaporating refrigerants, many predictive techniques for calculating coefficients of evaporation are listed in Table 2 in Chapter 4 of the 2005 *ASHRAE Handbook—Fundamentals*. The most verified predictive technique is the Shah correlation (Shah 1976, 1982). A series of tests is specified in ARI *Standard* 410 for obtaining heat transfer data for direct-expansion refrigerants inside tubes of a given diameter.

ASHRAE *Standard* 33 specifies laboratory apparatus and instrumentation, including procedure and operating criteria for conducting tests on representative coil prototypes to obtain basic performance data. Procedures are available in ARI *Standard* 410 for reducing these test data to the performance parameters necessary to rate a line or lines of various air coils. This information is available from various coil manufacturers for use in selecting ARI *Standard* 410 certified coils.

The following example illustrates a method for selecting coil size, row depth, and performance data to satisfy specified job requirements. The application is for typical cooling and dehumidifying coil selection under conditions in which a part of the coil surface on the entering air side operates dry, with the remaining surface wet with condensing moisture. Figure 9 shows the thermal diagram, dry-wet boundary conditions, and terminology used in the problem solution.

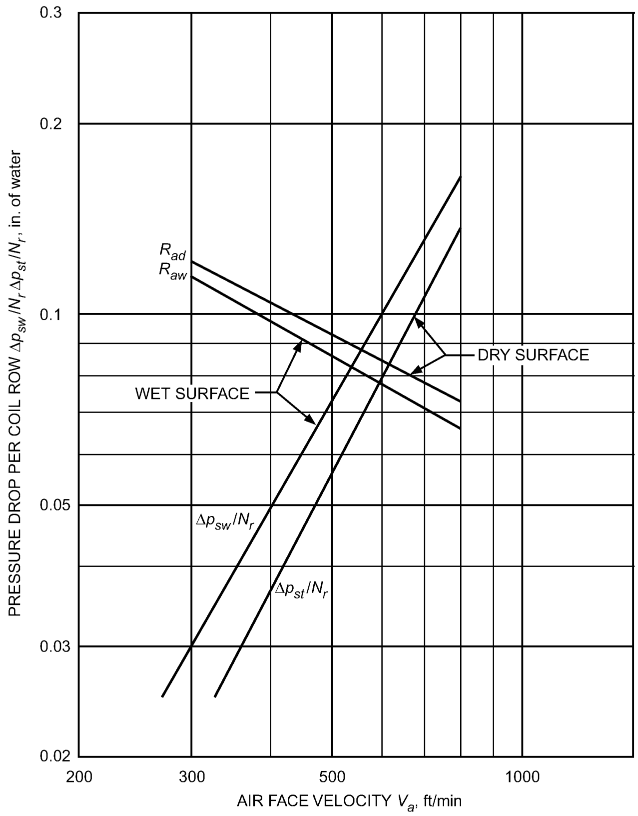

Fig. 12 Typical Air-Side Application Rating Data Determined Experimentally for Cooling and Dehumidifying Water Coils

Example 3. Standard air flowing at a mass rate equivalent of 6700 cfm enters a coil at 80°F db (t_{a1}) and 67°F wb (t'_{a1}). The air is to be cooled to 56°F leaving wet-bulb temperature t'_{a2} using 40 gpm of chilled water supplied to the coil at an entering temperature t_{r1} of 44°F, in thermal counterflow arrangement. Assume a standard coil air face velocity of V_a = 558 fpm and a clean, nonfouled, finned-tube heat transfer surface in the coil core, for which the following physical and performance data (such as illustrated in Figures 11 and 12) can be predetermined:

B = surface ratio = 25.9
c_p = 0.243 Btu/lb·°F
F_s = (external surface area)/(face area)(row deep) = 32.4
f_r = 750 Btu/h·ft^2·°F
$\Delta p_{sd}/N_r$ = 0.165 in. of water/number of coil rows, dry surface
$\Delta p_{sw}/N_r$ = 0.27 in. of water/number of coil rows, wet surface
R_{ad} = 0.073°F·ft^2·h/Btu
R_{aw} = 0.066°F·ft^2·h/Btu
R_{md} = 0.021°F·ft^2·h/Btu
R_{mw} = 0.0195°F·ft^2·h/Btu
ρ_a = 0.075 lb/ft^3
ρ_w = 8.34 lb/gal

Referring to Figure 9 for the symbols and typical diagram for applications in which only a part of the coil surface operates wet, determine (1) coil face area A_a, (2) total refrigeration load q_t, (3) leaving coolant temperature t_{r2}, (4) dry-wet boundary conditions, (5) heat transfer surface area required for dry A_d and wet A_w sections of the coil core, (6) leaving air dry-bulb temperature t_{a2}, (7) total number N_{ri} of installed coil rows, and (8) dry Δp_{st} and wet Δp_{sw} coil air friction.

Solution: The psychrometric properties and enthalpies for dry and moist air are based on Figure 1 (ASHRAE Psychrometric Chart No. 1) and Tables 2 and 3 in Chapter 6 of the 2005 *ASHRAE Handbook—Fundamentals* as follows:

h_{a1} = 31.52 Btu/lb$_a$ h''_{a1} = 26.67 Btu/lb$_a$
W_1 = 0.0112 lb$_w$/lb$_a$ $*h_{a2}$ = 23.84 Btu/lb$_a$
t''_{a1} = 60.3°F $*W_2$ = 0.0095 lb$_w$/lb$_a$

*As an approximation, assume leaving air is saturated (i.e., $t_{a2} = t'_{a2}$).

Calculate coil face area required:

$$A_a = 6700/558 = 12 \text{ ft}^2$$

From Equation (28b), find condensate heat rejection:

$$h_{fw} = (0.0112 - 0.0095)(1)(56 - 32) = 0.04 \text{ Btu/lb}_a$$

Compute the total refrigeration load from the following equation:

$$q_t = 60\rho_a w_a[h_{a1} - (h_{a2} + h_{fw})]$$
$$= 60 \times 0.075 \times 6700[31.52 - (23.84 + 0.04)] = 230{,}000 \text{ Btu/h}$$

From Equation (3), calculate coolant temperature leaving coil:

$$t_{r2} = t_{r1} + q_t/w_r c_r$$
$$= 44 + 230{,}000/(40 \times 60 \times 8.34 \times 1.0) = 55.5°\text{F}$$

From Equation (19), determine coolant film thermal resistance:

$$R_r = 25.9/750 = 0.0345°\text{F} \cdot \text{ft}^2 \cdot \text{h/Btu}$$

Calculate the wet coil characteristic from Equation (23):

$$C = \frac{0.0195 + 0.0346}{0.243 \times 0.066} = 3.37 \text{ lb}_a \cdot °\text{F/Btu}$$

Calculate from Equation (13):

$$y = \frac{55.5 - 44}{31.52 - 23.84} = 1.50 \text{ lb}_a \cdot °\text{F/Btu}$$

The dry-wet boundary conditions are determined as follows: From Equation (14), the boundary airstream enthalpy is

$$h_{ab} = \frac{(60.3 - 55.5) + (1.50 \times 31.52) + (3.37 \times 26.67)}{3.37 + 1.50}$$
$$= 29.15 \text{ Btu/lb}_a$$

According to limit (2) under Equation (14), part of the coil surface on the entering air side will be operating dry, because $h_{a1} > 29.15 > h_{a2}$ (see Figure 8).

From Equation (16), the boundary airstream dry-bulb temperature is

$$t_{ab} = 80 - (31.52 - 29.15)/0.243 = 70.25°\text{F}$$

The boundary surface conditions are

$$t_{sb} = t''_{a1} = 60.3°\text{F} \quad \text{and} \quad h_{sb} = h''_{a1} = 26.67 \text{ Btu/lb}_a$$

From Equation (17), the boundary coolant temperature is

$$t_{rb} = 55.5 - 1.50 \times 0.243(80 - 70.25) = 51.9°\text{F}$$

The cooling load for the dry surface part of the coil is now calculated from Equation (22b):

$$q_{td} = 40 \times 60 \times 8.34 \times 1.0 \times (55.5 - 51.9) = 72{,}000 \text{ Btu/h}$$

From Equation (18), the overall thermal resistance for the dry surface section is

$$R_o = 0.073 + 0.021 + 0.0346 = 0.129°\text{F} \cdot \text{ft}^2 \cdot \text{h/Btu}$$

From Equation (20), the mean temperature difference between air dry bulb and coolant for the dry surface section is

$$\Delta t_m = \frac{(80 - 55.5) - (70.25 - 51.92)}{\ln[(80 - 55.5)/(70.25 - 51.92)]} = 21.3°\text{F}$$

The dry surface area required is calculated from Equation (21):

$$A_d = 72{,}000 \times 0.129/21.3 = 436 \text{ ft}^2$$

From Equation (30), the cooling load for the wet surface section of the coil is

$$q_{tw} = 230{,}000 - 72{,}000 = 158{,}000 \text{ Btu/h}$$

Knowing C, h_{a2}, and t_{r1}, the surface condition at the leaving air side of the coil is calculated by trial and error using Equation (24):

$$C = 3.37 = (t_{s2} - 44)/(23.84 - h_{s2})$$

The numerical values for t_{s2} and h_{s2} are then determined directly by using a surface temperature chart (as shown in Figure 8 or in Figure 9 of ARI *Standard* 410) and saturated air enthalpies from Table 2 in Chapter 6 of the 2005 *ASHRAE Handbook—Fundamentals*:

$$t_{s2} = 51.03°\text{F} \quad \text{and} \quad h_{s2} = 20.88 \text{ Btu/lb}_a$$

From Equation (25), the mean effective difference in air enthalpy between airstream and surface is

$$\Delta h_m = \frac{(29.15 - 26.67) - (23.84 - 20.88)}{\ln[(29.15 - 26.67)/(23.84 - 20.88)]} = 2.713 \text{ Btu/lb}_a$$

From Equation (27a), the wet surface area required is

$$A_w = 158{,}000 \times 0.066 \times 0.243/2.713 = 934 \text{ ft}^2$$

From Equation (29), the net total surface area requirement for the coil is then

$$A_o = 436 + 934 = 1370 \text{ ft}^2 \text{ external}$$

From Equation (31), the net air-side heat transfer exponent is

$$c = 1370/(60 \times 0.075 \times 6700 \times 0.243 \times 0.073) = 2.56$$

From Equation (33), the enthalpy of saturated air corresponding to the effective surface temperature is

$$h_{\bar{s}} = 31.52 - \frac{31.52 - 23.84}{1 - e^{-2.56}} = 23.20 \text{ Btu/lb}_a$$

The effective surface temperature that corresponds to $h_{\bar{s}}$ is then obtained from Table 2 in Chapter 6 of the 2005 *ASHRAE Handbook—Fundamentals* as $t_{\bar{s}} = 54.95°\text{F}$.

The leaving air dry-bulb temperature is calculated from Equation (34):

$$t_{a2} = 54.95 + e^{-2.56}(80 - 54.95) = 56.9°\text{F}$$

The air-side sensible heat ratio is then found from Equation (35):

$$\text{SHR} = \frac{0.243(80 - 56.9)}{31.52 - 23.84} = 0.731$$

From Equation (1b), the calculated coil row depth N_{rc} to match job requirements is

$$N_{rc} = A_o/A_aF_s = 1370/(12 \times 32.4) = 3.5 \text{ rows deep}$$

In most coil selection problems of this type, the initial calculated row depth to satisfy job requirements is usually a noninteger value. In many cases, there is sufficient flexibility in fluid flow rates and operating temperature levels to recalculate the required row depth of a given coil size to match an available integer row depth more closely. For this example, if the calculated row depth is $N_{rc} = 3.5$, and coils of three or four rows deep are commercially available, the coil face area, operating conditions, and fluid flow rates and/or velocities could possibly be changed to recalculate a coil depth close to either three or four rows. Although core tube circuitry has limited possibilities on odd (e.g., three or five) row coils, alternative coil selections for the same job are often made desirable by trading off coil face size for row depth.

Most coil manufacturers have computer programs to run the iterations needed to predict operating values for specific coil performance requirements. The next highest integral row depth than computed is then selected for a commercially available coil with an even number of circuits, same end connected. For this example, assume that the initial coil selection requiring 3.5 rows deep is sufficiently refined that no recalculation is necessary, and that a 4-row coil with 4-pass coil circuitry is available. Thus, the installed row depth N_{ri} is

$$N_{ri} = 4 \text{ rows deep}$$

The amount of heat transfer surface area installed is

$$A_{oi} = A_aF_sN_{ri} = 12 \times 32.4 \times 4 = 1555 \text{ ft}^2 \text{ external}$$

The completely dry and completely wetted air-side frictions are, respectively,

$$\Delta p_{sd} = (A_d/A_o)(\Delta p_{sd}/N_{ri})N_{ri} = (436/1370) \times 0.165 \times 4$$
$$= 0.21 \text{ in. of water}$$

and

$$\Delta p_{sw} = (A_w/A_o)(\Delta p_{sw}/N_{ri})N_{ri} = (934/1370) \times 0.27 \times 4$$
$$= 0.74 \text{ in. of water}$$

$$\Delta p_{st} = (A_{oi}/A_o)\Delta p_{sd} + \Delta p_{sw} = (1555/1370) \times (0.21 + 0.74)$$
$$= 1.08 \text{ in. of water total}$$

A more realistic Δp estimate of a coil operating at >70% wetted surface and a velocity > 400 fpm would be to consider the entire surface as wetted. Therefore, $0.27 \times 4 = 1.08$ in. of water would be the coil's operating air-side static pressure.

In summary,

A_a	=	12 ft² coil face area
N_{ri}	=	4 rows installed coil depth
A_{oi}	=	1555 ft² installed heat transfer surface area
A_o	=	1370 ft² required heat transfer surface area
q_t	=	230,000 Btu/h total refrigeration load
t_{r2}	=	55.5°F leaving coolant temperature
t_{a2}	=	56.9°F leaving air dry-bulb temperature
SHR	=	0.731 air sensible heat ratio
Δp_{sw}	=	0.74 in. of water wet-coil surface air friction
Δp_{sd}	=	0.21 in. of water dry-coil surface air friction
Δp_{st}	=	1.08 in. of water total coil surface air friction

DETERMINING REFRIGERATION LOAD

The following calculation of refrigeration load distinguishes between the true sensible and latent heat loss of the air, which is accurate within the data's limitations. This division will not correspond to load determination obtained from approximate factors or constants.

The total refrigeration load q_t of a cooling and dehumidifying coil (or air washer) per unit mass of dry air is indicated in Figure 13 and consists of the following components:

- The sensible heat q_s removed from the dry air and moisture in cooling from entering temperature t_1 to leaving temperature t_2
- The latent heat q_e removed to condense the moisture at the dewpoint temperature t_4 of the entering air

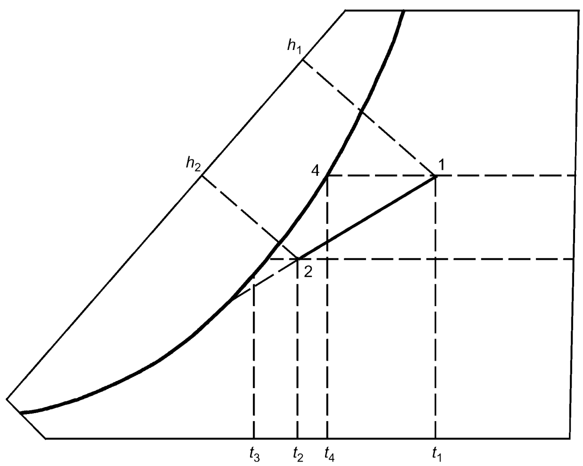

Fig. 13 Psychrometric Performance of Cooling and Dehumidifying Coil

- The heat q_w removed to further cool the condensate from its dew point t_4 to its leaving condensate temperature t_3

The preceding components are related by the following equation:

$$q_t = q_s + q_e + q_w \tag{36}$$

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

$$q_t = (h_1 - h_2) - (W_1 - W_2)h_{w3} \tag{37}$$

where

h_1 and h_2 = enthalpy of air at points 1 and 2, respectively
W_1 and W_2 = humidity ratio at points 1 and 2, respectively
h_{w3} = enthalpy of saturated liquid at final temperature t_3

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

Latent heat may be found from

$$q_e = (W_1 - W_2)h_{fg4} \tag{38}$$

where

h_{fg4} = enthalpy representing latent heat of water vapor at condensing temperature t_4

Sensible heat may be shown to be

$$q_s + q_w = (h_1 - h_2) - (W_1 - W_2)h_{g4} + (W_1 - W_2)(h_{w4} - h_{w3}) \tag{39a}$$

or

$$q_s + q_w = (h_1 - h_2) - (W_1 - W_2)(h_{fg4} + h_{w3}) \tag{39b}$$

where

h_{g4} = $h_{fg4} + h_{w4}$ = enthalpy of saturated water vapor at condensing temperature t_4
h_{w4} = enthalpy of saturated liquid at condensing temperature t_4

The last term in Equation (39a) is the heat of subcooling the condensate from t_4 to its final temperature t_3. Then,

$$q_w = (W_1 - W_2)(h_{w4} - h_{w3}) \tag{40}$$

The final condensate temperature t_3 leaving the system is subject to substantial variations, depending on the method of coil installation, as affected by coil face orientation, airflow direction, and air duct insulation. In practice, t_3 is frequently the same as the leaving wet-bulb temperature. Within the normal air-conditioning range, precise values of t_3 are not necessary because the heat q_t of condensate removed from the air usually represents about 0.5 to 1.5% of the total refrigeration cooling load.

Example 4. Air enters a coil at 90°F db, 75°F wb; it leaves at 61°F db, 58°F wb; leaving water temperature is assumed to be 54°F, which is between the leaving air dew point and coil surface temperature. Find the total, latent, and sensible cooling loads on the coil with air at standard barometric pressure.

Solution: Using Figure 1 (or the indicated equations) from Chapter 6 of the 2005 *ASHRAE Handbook—Fundamentals*,

h_1	= 38.37 Btu/lb$_a$	(32)
h_2	= 25.06 Btu/lb$_a$	(32)
W_1	= 0.01523 lb$_w$/lb$_a$	(35)
W_2	= 0.00958 lb$_w$/lb$_a$	(35)
t_4	= 69.04°F dew point of entering air	(39)
h_{w4}	= 37.04 Btu/lb	(34)
h_{w3}	= 22.00 Btu/lb	(34)
h_{g4}	= 1091.66 Btu/lb	(31)
h_{fg4}	= 1054.61 Btu/lb	(31)

From Equation (37), the total heat is

$$q_t = (38.37 - 25.06) - (0.01523 - 0.00958)22.00 = 13.19 \text{ Btu/lb}_a$$

From Equation (38), the latent heat is

$$q_e = (0.01523 - 0.00958)1054.61 = 5.96 \text{ Btu/lb}_a$$

The sensible heat is therefore

$$q_s + q_w = q_t - q_e = 13.19 - 5.96 = 7.23 \text{ Btu/lb}_a$$

The sensible heat may be computed from Equation (39a) as

$$q_s + q_w = (38.37 - 25.06) - (0.01523 - 0.00958)1091.66 + (0.01523 - 0.00958)(37.04 - 22.00) = 7.22 \text{ Btu/lb}_a$$

The same value is found using Equation (39b). The subcooling of the condensate as a part of the sensible heat is indicated by the last term of the equation, 0.08 Btu/lb$_a$.

MAINTENANCE

If the coil is to deliver its full cooling capacity, both its internal and external surfaces must be clean. The tubes generally stay clean in pressurized water or brine systems. Tube surfaces can be cleaned in a number of ways, but are often washed with low-pressure water spray and mild detergent. Water coils should be completely drained if freezing is possible. When coils use built-up system refrigerant evaporators, oil can accumulate. Check and drain oil occasionally, and check for leaks and refrigerant dryness.

Air Side. The best maintenance for the outside finned area is consistent inspection and service of inlet air filters. Surface cleaning of the coil with pressurized hot water and a mild detergent should be done only when necessary (primarily when a blockage occurs under severe fin-surface-fouling service conditions, or bacterial growth is seen or suspected). Pressurized cleaning is more thorough if done first from the air exit side of the coil and then from the air entry side. Foaming chemical sprays and washes should be used instead of high pressure on fragile fins, or when fin density is too restrictive to allow proper in-depth cleaning with pressurized water spray. In all cases, limit spray water temperature to below 150°F on evaporator coils containing refrigerant. In cases of marked neglect or heavy-duty use (especially in restaurants where grease and dirt have accumulated) coils must sometimes be removed and the accumulation washed off with steam, compressed air and water, or hot water.

The surfaces can also be brushed and vacuumed. Best practice is to inspect and service the filters frequently. Also, condensate drain pan(s) and their drain lines, including open drain areas, should be kept clean and clear at all times.

Water Side. The best service for the inside tube coil surface is keeping the circulated fluid (water or glycol) free of sediment, corrosive products, and biological growth. Maintaining proper circulated water chemistry and velocity and filtering out solids should minimize the water-side fouling factor. If large amounts of scale form when untreated water is used as coolant, chemical or mechanical (rod) cleaning of internal surfaces at frequent intervals is necessary. A properly maintained chilled-water system using a glycol solution as the circulated fluid is not considered to ever have water-side fouling, as such, but glycol solutions must be analyzed seasonally to determine alkalinity, percent concentration, and corrosion inhibitor condition. Consult a glycol expert or the manufacturer's agent for detailed recommendations on proper use and control of glycol solutions.

Refrigerant Side. Moisture content of the refrigerant should be checked yearly, and acidity of the compressor oil as often as monthly. For built-up direct-expansion (DX) systems, normal is ≤50 ppm of moisture for systems with mineral oil, and ≤100 ppm for some refrigerant types in systems with polyol ester (POE) compressor oil. The actual values are set by the compressor manufacturer, and should be checked and verified during operation by moisture-indicating sight-glass viewing, and yearly by laboratory sample analysis reports. Depending on temperature and velocity, excessive moisture coming to the cooling coil through refrigerant might cause internal freeze-up around the coil's expansion valve. Generally, moisture in a system contributes to formation of acids, sludge, copper plating, and corrosion. Oil breakdown (by excessive compressor overheat) can form organic, hydrofluoric, and hydrochloric acids in the lubricant oil, all of which can corrode copper.

The refrigerant must be of high quality and meet ARI *Standard* 700 purity requirements. Application design ratings of the coil or coil bank are based on purity and dryness. This is a primary requirement of refrigerant evaporator coil maintenance.

DX coil replacement often coincides with refrigerant change-out to a chlorine-free refrigerant. Special care should be taken to ensure that the system is retrofitted in accordance with the compressor and refrigerant manufacturers' written procedures.

SYMBOLS

A_a = coil face or frontal area, ft²
A_d = dry external surface area, ft²
A_o = total external surface area, ft²
A_p = exposed external prime surface area, ft²
A_s = external secondary surface area, ft²
A_w = wet external surface area, ft²
B = ratio of external to internal surface area, dimensionless
C = coil characteristic as defined in Equations (12) and (23), $lb_a \cdot °F/Btu$
c = heat transfer exponent, or NTU_a, as defined in Equations (31) and (32), dimensionless
c_o = heat transfer exponent, as defined in Equation (7d), dimensionless
c_p = specific heat of humid air = 0.243 $Btu/lb_a \cdot °F$ for cooling coils
c_{pw} = specific heat of water = 1.0 $Btu/lb_w \cdot °F$
c_r = specific heat of nonvolatile coolant, $Btu/lb_a \cdot °F$
D_i = tube inside diameter, in.
D_o = tube outside diameter, in.
E_a = air-side effectiveness defined in Equation (7b), dimensionless
F_s = coil core surface area parameter = (external surface area)/(face area) (no. of rows deep)
f = convection heat transfer coefficient, $Btu/h \cdot ft^2 \cdot °F$
h = air enthalpy (actual in airstream or saturation value at surface temperature), Btu/lb_a
Δh_m = mean effective difference of air enthalpy, as defined in Equation (25), Btu/lb_a
k = thermal conductivity of tube material, $Btu/h \cdot ft \cdot °F$
M = ratio of nonvolatile coolant-to-air temperature changes for sensible heat cooling coils, as defined in Equation (7e), dimensionless
m'' = rate of change of air enthalpy at saturation with air temperature, $Btu/lb \cdot °F$
N_r = number of coil rows deep in airflow direction, dimensionless
Δp_{sd} = isothermal dry surface air-side pressure drop at standard conditions (70°F, 29.92 in. Hg), in. of water
Δp_{sw} = wet surface air-side pressure drop at standard conditions (70°F, 29.92 in. Hg), in. of water
q = heat transfer capacity, Btu/h
q_e = latent heat removed from entering air to condense moisture, Btu/lb_a
q_s = sensible heat removed from entering air, Btu/lb_a
q_t = total refrigeration load of cooling and dehumidifying coil, Btu/lb_a
q_w = sensible heat removed from condensate to cool it to leaving temperature, Btu/lb_a
R = thermal resistance, referred to external area A_o, $h \cdot °F \cdot ft^2/Btu$
SHR = ratio of air sensible heat to air total heat, dimensionless
t = temperature, °F
Δt_m = mean effective temperature difference, air dry bulb to coolant temperature, °F
Δt_{ms} = mean effective temperature difference, surface-to-coolant, °F
$\Delta t_{\bar{m}}$ = mean effective temperature difference, air dry bulb to effective surface temperature $t_{\bar{s}}$, °F
U_o = overall sensible heat transfer coefficient, $Btu/h \cdot ft^2 \cdot °F$
V_a = coil air face velocity at 70°F, fpm
W = air humidity ratio, pounds of water per pound of air
w = mass flow rate, lb/h
y = ratio of nonvolatile coolant temperature rise to airstream enthalpy drop, as defined in Equation (13), $lb_a \cdot °F/Btu$
η = fin effectiveness, as defined in Equation (6), dimensionless
ρ_a = air density = 0.075 lb/ft³ at 70°F at sea level

Superscripts

$'$ = wet bulb
$''$ = dew point

Subscripts

1 = condition entering coil
2 = condition leaving coil
a = airstream
ab = air, dry-wet boundary
ad = dry air
aw = wet air
b = dry-wet surface boundary
d = dry surface
e = latent
f = fin (with R); saturated liquid water (with h)
g = saturated water vapor
i = installed, selected (with A_o, N_r)
m = metal (with R) and mean (with other symbols)
md = dry metal
mw = wet metal
o = overall (except for A)
r = coolant
rb = coolant dry-wet boundary
s = surface (with d, t, and w) and saturated (with h)
$\bar{s}$ = effective surface
sb = surface dry-wet boundary
t = tube (with R) and total (with s and q)
td = total heat capacity, dry surface
tw = total heat capacity, wet surface
w = water (with ρ), condensate (with h and subscript number), and wet surface (with other symbols)

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