C2.5 Air-Cooled Heat Exchanger Construction Practices

C2.5.1 Introduction

An air-cooled heat exchanger (ACHE) initially costs more than a water-cooled one, such as shell-and-tube or gasketed-plate. Over the life of the unit, however, the added cost of providing a continuous supply of treated cooling water drives up the water-cooled heat exchanger's operating expense. Consequently, the ACHE might represent the better long-term investment.

Section C2.5 examines various air-cooled heat exchanger configurations and summarizes current construction and design practices described in Reports AC-2 [1], AC-6 [2], and ACD-2.1 [3]. Results from our forced-draft air-cooler model tests [4] are also presented. Later sections discuss the effects of fan height above ground, plenum depth, fan intake blockage, multiple fan interactions, tapered plenum, fan rings, fan guards, louvers, steam coils, and hail screens. Design methods that reduce or eliminate back-flow and inert blanketing (detailed in AC-3 [5] and AC-4 [6]) are summarized.

The mechanical design of an air-cooled heat exchanger relies on principles of pressure vessel design, structural engineering, and rotating equipment theory. The following sections summarize the influence of these principles on current ACHE design methods. For detailed information, users are directed to relevant national and international design codes for such equipment.

C2.5.2 Description of Air-Cooled Heat Exchangers

Certain terms and definitions, common among ACHE designers, are used throughout this section:

- tube bundle – assembly of headers, tubes, and frames
- bay – one or more tube bundles served by two or more fans, complete with structure, plenum, and other attendant equipment
- unit – one or more tube bundles in one or more bays performing a single service
- bank – one or more bays that include one or more units arranged in a continuous structure
- forced draft – exchanger designed with tube bundles on the discharge side of the fan

C2.5.2 Description of Air-Cooled Heat Exchangers, continued

- induced draft – exchanger designed with tube bundles on the suction side of the fan

Following are the main components of an air-cooled heat exchanger:

- one or more bundles of heat transfer surface
- air-moving device such as a fan, blower, or stack
- driver and power transmission to mechanically rotate the fan
- plenum between the bundle (or bundles) and the air moving device
- support structure high enough to allow air to enter beneath the air-cooled heat exchanger at a reasonable approach velocity

These components are sometimes included in the ACHE design:

- header and fan maintenance walkways and ladders to the ground
- louvers for process outlet temperature control
- chamber recirculation duct and/or steam coils for cold weather protection against freezing or solidification of high pour-point materials
- variable pitch fan hub and/or variable speed fan motors for temperature control and power savings
- hail screens to protect the bundle from damage

C2.5.3 Air-Cooled Heat Exchanger Configurations

In some situations, the choice of heat exchanger type is critical to proper plant operation; the project engineer must therefore understand advantages and drawbacks associated with various configurations.

C2.5.3.1 Forced-Draft, Horizontal Bundle

Horizontal arrangements are the most common forced-draft design. The size of shop-erected units is limited by transportation restrictions; field-erected units are larger, their size limited only by practicality. Units with average length-to-width ratios ranging from 2–2.5 to 1 require a two-fan design. Large process coolers have two or more fans.

C2.5.3.1 Forced-Draft, Horizontal Bundle, continued

Forced-draft air-cooled heat exchangers with horizontal bundles have these advantages over induced-draft ACHEs:

- Less power is needed to convey air because fans are located in the cool airstream below the bundle.
- Maintenance is easier because fan drives are located below the unit.
- The construction material is not critical because fans are unlikely to overheat unless near a very high temperature bundle or in a recirculation cabin.
- Bundles are located above the plenum chamber, which simplifies assembly of the structure. Disassembly is usually not required to remove bundles for cleaning or repair.

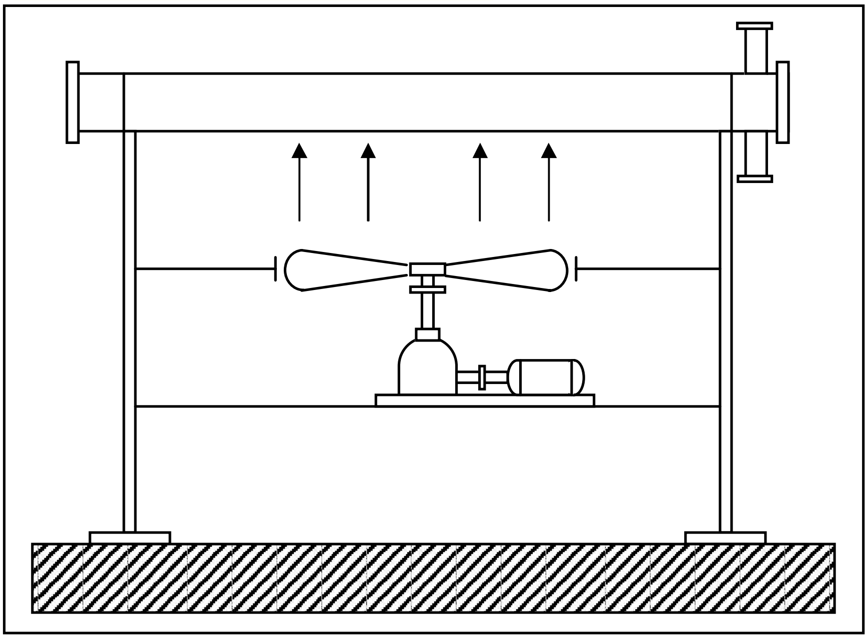

Figure C2.5-1. Forced-draft, horizontal configuration.

This type of design also has several disadvantages:

- Unless the unit is grade-mounted, underslung walkways are required for motor and fan access.
- The velocity of air escaping from the bundle's top is low—typically 500–700 ft/min (2.5–3.5 m/s)—making the unit susceptible to crosswind effects and inducing external recirculation around the cooler. This problem is accentuated by proximity to tall structures or to other units that are not part of the same continuous bank. Anti-recirculation fences may have to be fitted at considerable expense.

C2.5.3.1 Forced-Draft, Horizontal Bundle, continued

- Good airflow distribution is more difficult to achieve than with induced-draft exchangers.
- The bundles are exposed to solar radiation, which increases the heat load. For most cases, the increase in heat load is small (< 2%) and can be neglected. However, for cases where the effective mean temperature difference is low (< 5.6 °C (10 °F)) and the tubeside heat transfer coefficient is low (for example, in laminar single-phase flow), the solar radiation can increase the duty more than 5% and should be included in the performance analysis.

C2.5.3.2 Induced-Draft, Horizontal Bundle

Min. Temp. approach
Forced draft Induced draft
12 C 8 C

Horizontal induced-draft units, often designed for processes requiring considerable cooling surface, are usually multiple-bay installations.

Advantages typical of this design follow:

- The unit is less susceptible to crosswind because the velocity of air discharging from the fan can reach 32.8 ft/s (10 m/s).
- Cooling air is less likely to recirculate than in other designs.
- The plenum chamber, whether a hood or a flat deck, protects against sudden performance surges caused by rain or hail; it also reduces the effects of solar radiation on the bundle, making horizontal induced-draft units popular in the Middle East.
- Acting as a chimney, the plenum chamber and fan ring provide a higher heat rejection under fan failure conditions.

For close temperature control; i.e. induced draft when +/- 3 C control is required.

C2.5.3.2 Induced-Draft, Horizontal Bundle, continued

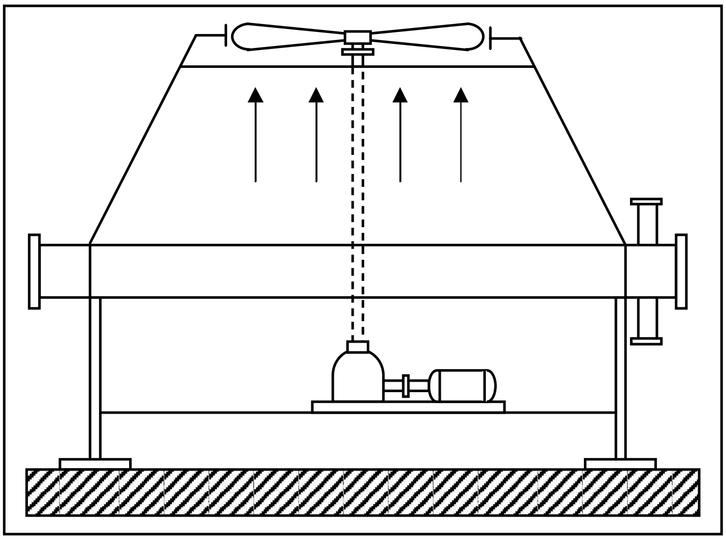

Figure C2.5-2. Induced-draft, horizontal configuration

A few disadvantages are also ascribed to this configuration:

- Except for very low process temperatures (below 158 ˚F (70 ˚C)), the gearbox or belt drive system cannot be mounted in the hot airstream.
- Except for those with remote actuation, all autovariable fans have low maximum operating temperatures and are unsuitable for mounting in hot airstreams.
- All fans made of or containing combustible materials (e.g., plastic, rubber) have low temperature limits. The unit must be rated at the maximum process temperature with the motor off to ensure the fan's suitability for service.
- With the fan operating in a warm airstream, the unit's power consumption will be higher for a given thermal performance.

C2.5.3.3 A-Frame Configuration

Power plants use this design almost exclusively for steam condensation. The A-frame's cooler bundles (usually sloped between 45 and 60 degrees from the horizontal) are mounted on a triangular frame with forced-draft fans below.

C2.5.3.3 A-Frame Configuration, continued

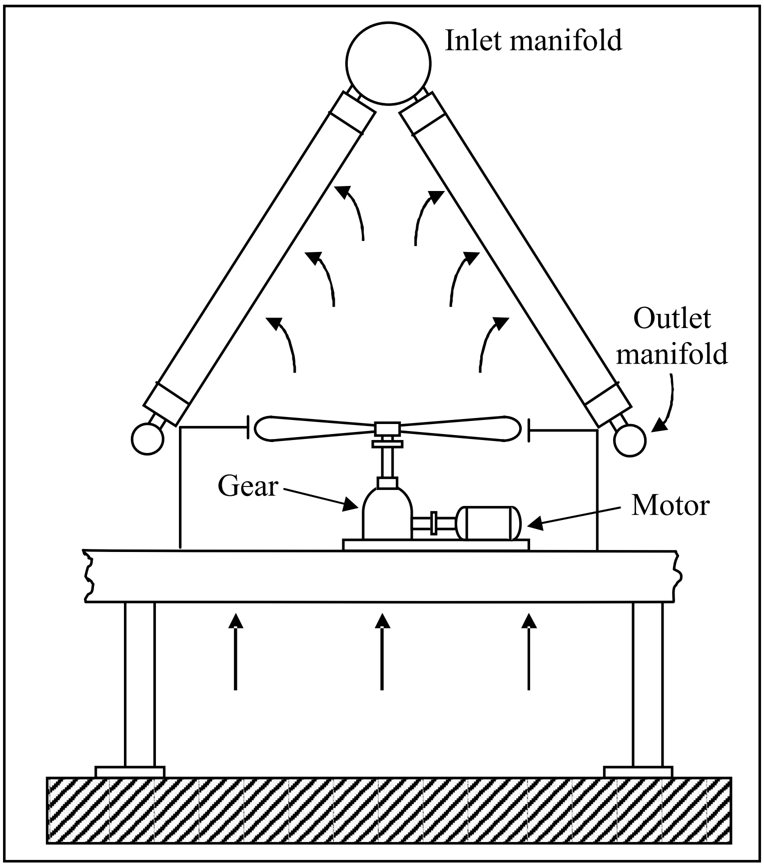

Figure C2.5-3. A-frame configuration

The A-frame design has these advantages:

- The configuration requires 65 – 70 percent less plot area than a conventional induced or forced-draft layout.
- The design is the basis of several patented non-freezing systems related to vacuum steam condensers (see Report AC-4 [6]).
- The A-frame, with inclined tube bundles that facilitate condensate drainage, is ideally suited for condensing applications. The common header at the apex of the A allows for optimum steam distribution with minimum head loss, an important consideration in vacuum steam condensers.

C2.5.3.3 A-Frame Configuration, continued

Conversely, the design has these disadvantages:

- A-frames are more prone to hot air recirculation than other designs.
- Susceptible to crosswind effects in bad weather, the A-frame may require a preventative wind/anti-recirculation fence.
- When the tubes are not horizontal, multipass bundles cannot be used for condensing services and bundles in series may be required.

C2.5.3.4 Vertical Coolers

As Figures C2.5-4 and C2.5-5 show, vertical coolers with forced or induced draft have tube bundles at or near vertical. Vertical units are generally incorporated as part of a packaged system.

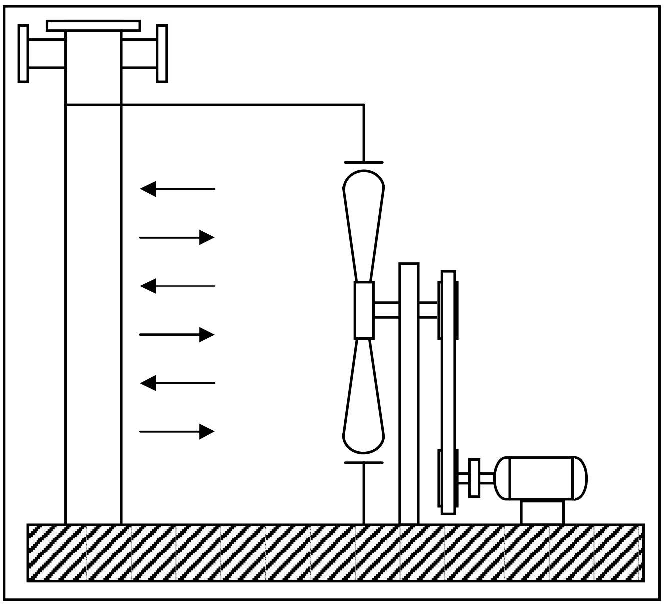

Figure C2.5-4. Vertical configuration, forced or induced draft

C2.5.3.4 Vertical Coolers, continued

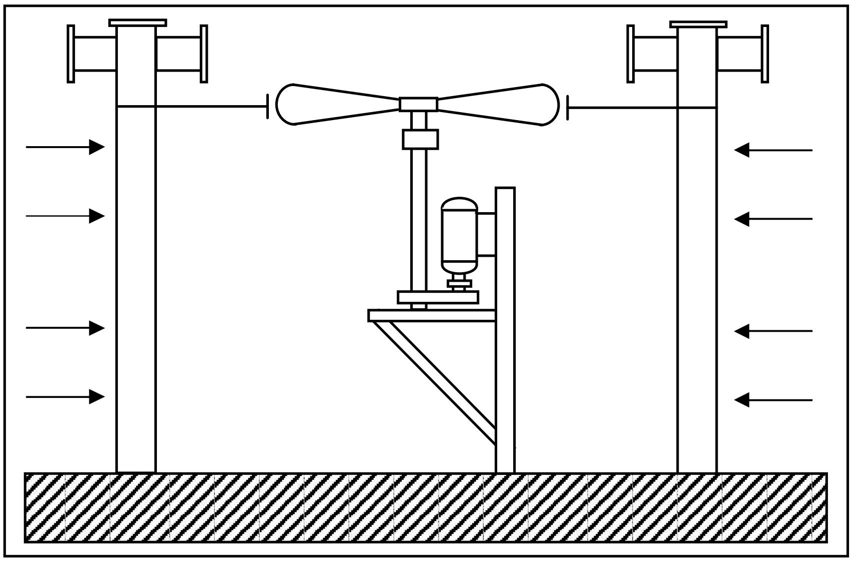

Figure C2.5-5. Vertical configuration, induced draft (box construction)

The vertical cooler has two significant advantages:

- Vertical coolers can be used in small areas.
- Units of this design, such as column-top reflux condensers, can be mounted directly on columns.

This design also has these disadvantages:

- The design is susceptible to crosswind effects.
- The use of multipass condensers is not practical.

C2.5.4 Tube Bundles

An air-cooler tube bundle is composed of side frames, headers, tube supports, and the tube field.

The tubes exposed to the passage of air usually have fins that form an extended surface. This surface compensates for the low film coefficient of air at atmospheric pressure and the usually low velocities across the bundle. The base tube is typically round and composed of material suited for such process considerations as corrosion, pressure, and temperature limitations. Whether helical or plate, the fins are usually made of aluminum to improve thermal conductivity and lessen fabrication costs. Very high temperature applications require steel fins, however.

C2.5.4 Tube Bundles, continued

Finned tubes are made in one of several ways:

- Fins are extruded from an aluminum jacket on the tube.
- A ribbon of aluminum is wrapped around the tube in a precut helical groove, the edge of which is peened back to secure the strip.
- Tubes are inserted through openings in closely spaced aluminum plates and then expanded for close fit.
- Fins are welded, brazed, or soldered to the tubes.

Reports AC-6 [2] and ESG-11 [7] provide additional details.

Cut or serrated fins interrupt the air boundary layer, increasing turbulence and producing a higher air film coefficient with only a slight increase in air pressure loss. For information on serrated finned tubes, refer to ESG-13 [8].

Important factors to consider when selecting finned tubes are listed below:

- heat transfer characteristics
- pressure drop characteristics
- cost
- temperature limitations of construction material
- initial effectiveness of the contact bond between tube and fin and subsequent possibility of relaxation [7]
- possibility of corrosion of the tube at the base under the fin

The extruded fin provides a reliable bond and a complete sheath, protecting the full length of the base tube against atmospheric corrosion.

Tubes are 6–50 ft (1.83–15.25 m) long with diameters from 5/8 to 6 in. (15.9 to 152.4 mm), the most common being 1 in. (25.4 mm). Fins are commonly helical, have 8–12 fins per inch (315–472 fins per meter), and range between 0.010 and 0.035 in. (0.254 and 0.889 mm) thick. The ratio of extended to prime surface varies from 7 to 25.

Rectangular bundles typically consist of 2 to 10 rows of finned tubes arranged on a triangular pitch of approximately 2.5 tube diameters. The tubes are rolled or welded into the tubesheets of a pair of headers.

C2.5.4 Tube Bundles, continued

The three basic header types are box, removable cover, and manifold. For headers without a removable cover, threaded holes opposite each tube end allow maintenance of the tubes; a plug is screwed into each hole. Bolted cover plates are removable for quick and easy access to applications that have fouling characteristics. Partitions welded in the headers provide multiple tubepasses to increase fluid velocities, provide flow as near countercurrent as possible, and improve the mean temperature differences [6]. Partitions or stiffeners (partitions with flow openings) also act as structural stays.

Horizontally split headers may be needed to accommodate differential tube expansion in applications with high fluid temperature differences per pass.

Any number of bundles can be combined in a single unit with one set of fans.

C2.5.5 Axial Flow Fans

ACHEs commonly employ an axial flow propeller fan. Fan coverage, the ratio of the fan's projected area to the bundles' face area, should be maintained at approximately 40 percent to prevent poor air distribution across the bundle face. HTRI recommends fan placement that allows a dispersion angle of no more than 45 degrees at the bundle centerlines [2].

Fan diameters may be as large as 60 ft (18.3 m). Fans can have from 2 to 20 blades made from wood, steel, solid or hollow aluminum, or fiberglass-reinforced plastic— the latter being the most popular. The blade edge can be straight or contoured. The most efficient blade is wide at the center, has a slight twist, and tapers near the edge. Blade pitch can be fixed or adjustable. Except for those with diameters of less than 5 ft (1.53 m), most air-cooled heat exchangers have fans with adjustable blade pitch. The pitch can be changed manually or automatically; the adjusting mechanism is pneumatic on smaller fans and electrical on larger sizes.

C2.5.5.1 Typical Fan Performance Curve

A propeller fan with a typical performance curve (see Figure C2.5-6) can furnish high volumes of air at low to moderate static pressure. Such fans have a large outlet area, moderately low outlet velocities, and efficiencies reaching 80 percent. Where noise is a factor, blade tip speeds range from about 5500 to 9000 ft/min (28 to 46 m/s); where noise is not a consideration, speeds can be up to 13000 ft/min (66 m/s). The American

C2.5.5.1 Typical Fan Performance Curve, continued

Petroleum Institute Standard recommends maintaining fan speed below 12000 ft/min (61 m/s), except for direct-driven fans. That type can have tip speeds up to 16000 ft/min (81 m/s), unless stricter limitations are required to maintain a specified noise level.

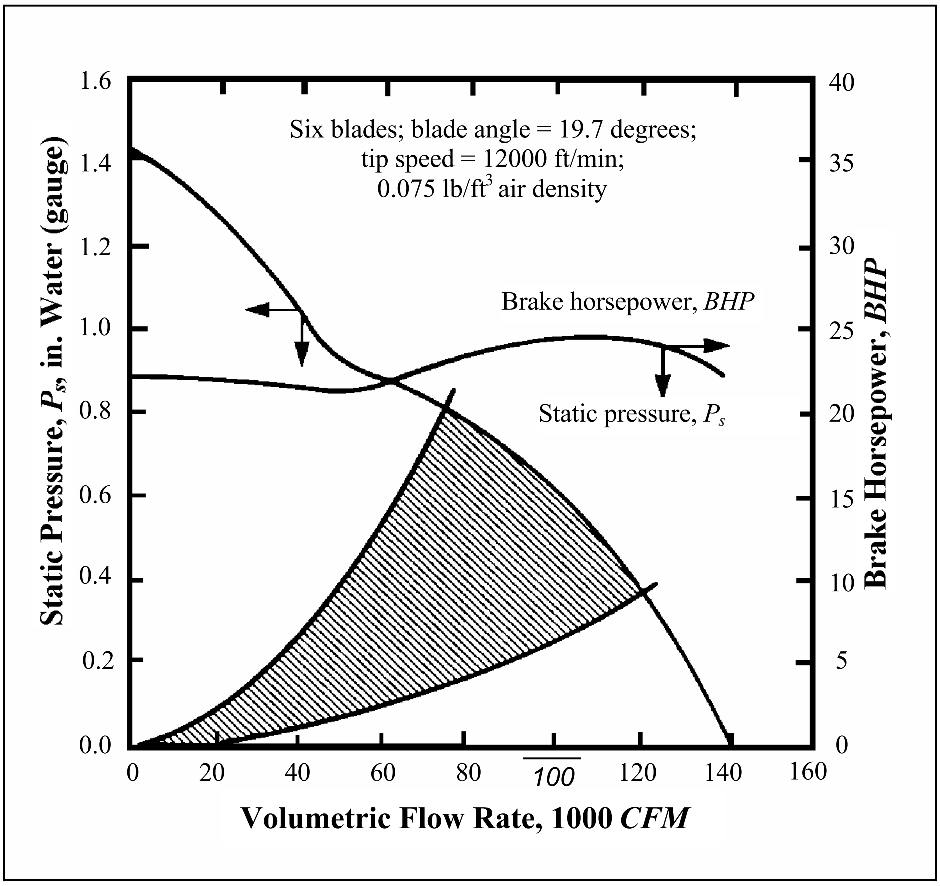

Figure C2.5-6. Performance curve of 8-ft OD fan

HTRI suggests a fan operating range between about 55 percent to 85 percent of air capacity. This range assures little variation in brake horsepower, and both static and total efficiency usually peak between these points. A reasonable preliminary estimate of total efficiency is 65 percent. The following empirical formula provides an estimate of horsepower required by the fan:

$$BHP = \frac{(ACFM)\,P_t}{4130}\frac{\rho}{\rho_s} \qquad \text{(C2.5-1)}$$

C2.5.5.2 Effect of Air Density

The propeller fan can be considered a constant volume machine at a constant speed and angle.

$$CFM_2 = CFM_1 \tag{C2.5-2}$$

$$BHP_2 = BHP_1\left(\frac{\rho_2}{\rho_1}\right) = BHP_1\left(\frac{T_1}{T_2}\right) \tag{C2.5-3}$$

$$(P_s)_2 = (P_s)_1\left(\frac{\rho_2}{\rho_1}\right) = (P_s)_1\left(\frac{T_1}{T_2}\right) \tag{C2.5-4}$$

C2.5.5.3 Effect of Speed Change

At a constant blade angle, the volumetric flow rate is directly proportional to the fan speed, the static pressure head is proportional to the fan speed squared, and the brake horsepower is proportional to the fan speed cubed.

$$CFM_2 = CFM_1\left(\frac{S_2}{S_1}\right) \tag{C2.5-5}$$

$$(P_s)_2 = (P_s)_1\left(\frac{S_2}{S_1}\right)^2 \tag{C2.5-6}$$

$$BHP_2 = BHP_1\left(\frac{S_2}{S_1}\right)^3 \tag{C2.5-7}$$

These well-documented laws assume that efficiency does not change along the fan curve for speeds typical in heat exchanger work.

Report AC-2 details such design effects as blade angle, number, and width [1].

C2.5.6 Plenum, Fan Deck, and Fan Ring Construction

Plenums, either box- or transition-type, are constructed of ribbed 14 gauge steel sheets, 0.083-in. (2.1-mm) minimum thickness.

A large deck or one made from welded plates requires bracing. A fabricated fan deck should support 50 lb/ft^2 (245 kg/m^2) and be constructed of 12 gauge steel, 0.109-in. (2.77-mm) minimum thickness.

C2.5.6 Plenum, Fan Deck, and Fan Ring Construction, continued

Transition plenums, used primarily in induced-draft designs, are more rigid than the box-type and require no additional fan deck.

Fan tip clearance can significantly affect fan performance. A diametral tip clearance of either 1.5 in. (38.1 mm) or 1 percent of the fan diameter, whichever is less, represents a reasonable compromise between optimum clearance and manufacturing capabilities. HTRI experiments (recorded in Report ACD-2.1 [3]) verify that sealing the gap between the fan blade tip and the fan ring with filler material improves ACHE performance. If plastic strips are used, the fan ring manufacturer's tolerances can be relaxed.

Fan rings must be concentric and accurately sized, their depth also affecting performance. Adding an inlet transition (for forced draft) improves fan performance.

C2.5.7 Motor-Fan Drives

An electric motor, steam turbine, gas or gasoline engine, or hydraulic motor provides power for the fan. The electric motor, by far the most popular choice, is typically a polyphase squirrel-cage induction type. Direct, V-belt, cog-belt, or gear drives transmit power.

- **Direct drive**

  With this system, the fan shaft connects directly to the driver, both having the same rpm. This system features fans with diameters of 5 ft (1.53 m) or less and drives of 5 hp (3.73 kW) or less.

- **V-belt drive**

  This system is used when the fan's rpm is less than the driver's. The drive design, the ratio of the fan sheave diameter to the motor sheave diameter, and the distance between sheave centers should be such that the belt-to-sheave angle exceeds 120 degrees. With smaller contact angles, power transmission per belt decreases markedly. For power above 50 hp (37.3 kW), cog-belt or gear drive should be used.

- **Gear drive**

  Gear drives are specified when hp or starting torque is too great for belt drives or when speed reduction is outside the range of a belt drive.

- **Cog-belt drive**

  Transmitting over 50 hp (37.3 kW), this arrangement, though noisy, can be an economical alternative to a gear drive.

C2.5.8 Air Flow in Forced-Draft

The methods presented in this section predict airflow degradation due to effects of fan height, inlet airflow restrictions, plenum depth, and fan rings and guards. Additional methods, developed for horizontal-type ACHEs, estimate recirculation flow when only one of three fans is operating. These methods were based primarily on HTRI's experimental isothermal study using a one-third-scale model of a three-fan commercial installation. Effects of heat transfer and air density change were not studied. Results obtained from these methods may vary, depending on the fan type; HTRI chose a high-efficiency state-of-the-art design for the experiment. Report AC-5 [4] provides detailed information on the following methods.

C2.5.8.1 Auxiliary Pressure Drop

Ideally, fan performance curves are used to design or rate airflow of an ACHE.

The total system pressure drop includes pressure drop due to the bundle and any auxiliary components, such as fan ring, fan guard, etc.

$$\Delta P_{sys} = \Delta P_b + \Delta P_{aux} \quad \text{(C2.5-8)}$$

Methods in Section B2 illustrate bundle pressure drop, ΔP_b. Bundle pressure loss, the major component of Equation (C2.5-8), dominates most ACHE designs. Pressure drop attributed to auxiliary items is described below.

C2.5.8.1.1 Fan Rings

Air cooler fans are typically enclosed in rings because proper ring design can greatly enhance fan performance. The correlation used to calculate the pressure drop due to fan rings is [1]

$$\Delta P_{ring} = \frac{K\rho_{be}V_{fan}^2}{2} \quad \text{(C2.5-9)}$$

Experimentally measured K-factor values are tabulated in Table C2.5-1 [1], with ring geometries defined in Figure C2.5-7.

C2.5.8.1.1 Fan Rings, continued

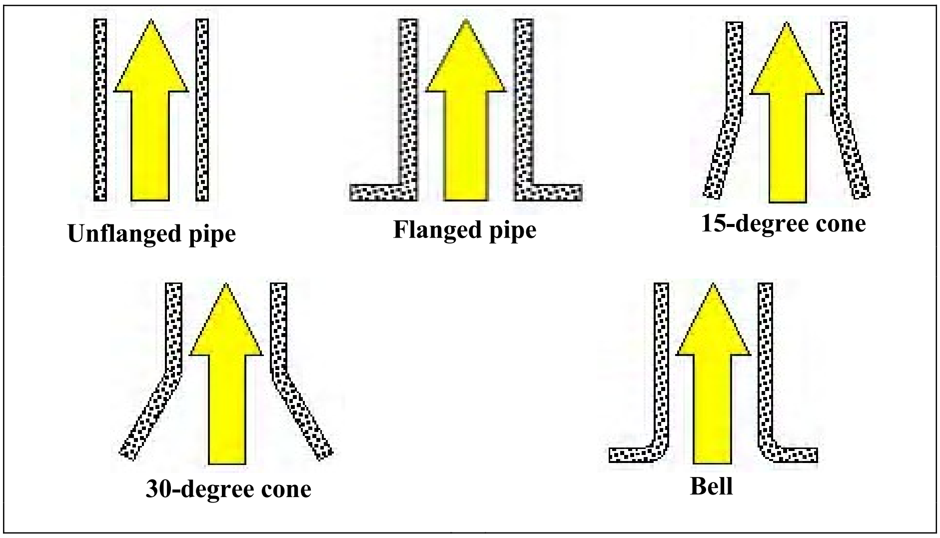

Figure C2.5-7. Fan ring geometries

Table C2.5-1. Fan Ring K-Factors (*K*)

Fan ring type	K-factor
Unflanged pipe	0.90
Flanged pipe	0.50
15-degree cone	0.13
30-degree cone	0.06
Smooth, well-rounded bell	0.05

C2.5.8.1.2 Fan Screens or Guards

Like fan rings, fan screens/guards are nearly ubiquitous in air cooler design because they protect both personnel and fans from damage. The equation used to predict pressure drop due to fan screens or guards is

$$\Delta P_{grd} = \frac{K \rho_{be} V_{fan}^2}{2} \qquad \text{(C2.5-10)}$$

However, the experimental determination of the fan screen/guard K-factors was based on velocities at the screen/guard. ***Xace*** uses the velocities at the fan, as demonstrated by Equation (C2.5-10). Thus, the two velocities are related through an intermediate K-factor, K_{int}:

C2.5.8.1.2 Fan Screens or Guards, continued

$$K = K_{\text{int}}\left[\left(\frac{100}{S_{nf}}\right)^2 - 1\right];\ 0 < K < 1 \qquad \text{(C2.5-11)}$$

Values of K_{int} depend on the fan ring and are listed in Table C2.5-2 [4].

Table C2.5-2. Fan Screen or Guard Intermediate K-Factors (K_{int})

Fan ring type	Intermediate K-factor
Unflanged pipe, forced draft	0.23
Unflanged pipe, induced draft	0.33
Unflanged cone, 15° taper	0.15
Unflanged cone, 30° taper	0.15
Unflanged dual cone, 45° into 15° taper	0.15
Smooth, well-rounded bell	0.13

C2.5.8.1.3 Ground Clearance

Insufficient ground clearance has a profound effect on the pressure drop incurred by an air cooler [1, 2, 4]. Low ground clearance results in more oblique airflow, essentially forming a *vena contracta* [1]. ***Xace*** does not require computation of the ground clearance contribution to pressure drop, even though it can be significant.

According to HTRI Reports AC-5 [4] and AC-7 [9], multiple factors influence the magnitude of the effects of ground clearance. Principal among these contributors are stagnation planes and airflow recirculation because they greatly disturb airflow uniformity. Any component that perturbs the airflow compounds both the threshold and the degree of ground clearance repercussions; experiments have shown that airflow can be decreased by as much as 70 percent.

The ground clearance pressure drop correlation is

$$\Delta P_{gc} = \frac{K\rho_{be}V_{be}^2}{2} \qquad \text{(C2.5-12)}$$

For ground clearance, the intermediate K-factor used is

C2.5.8.1.3 Ground Clearance, continued

$$K_{\text{int}} = \frac{H}{D} \tag{C2.5-13}$$

and the ground clearance K-factor correlations are [10]

$$K = 0.1448 - \frac{0.13682}{0.1} + \frac{0.209424}{0.1^2} = 19.719\ ;\ K_{int} \le 0.1 \tag{C2.5-14}$$

$$K = 0.1448 - \frac{0.13682}{K_{\text{int}}} + \frac{0.209424}{K_{\text{int}}^2}\ ;\ 0.1 < K_{int} \le 2 \tag{C2.5-15}$$

$$K = \left[\frac{0.1448 - \frac{0.13682}{2} + \frac{0.209424}{2^2}}{(2-2.5)}\right](K_{\text{int}} - 2) + \left(0.1448 - \frac{0.13682}{2} + \frac{0.209424}{2^2}\right);$$

$$2 < K_{int} \le 2.5 \tag{C2.5-16}$$

$$K = 0\ ;\ 2.5 < K_{int} \tag{C2.5-17}$$

These expressions for the K-factor assume that ground clearance loss coefficients are analogous to the loss coefficients experienced by a 90° smooth radius, die stamped, round elbow. The K-factor is limited to Equation (C2.5-14) at the high end and reaches zero when the ground clearance to fan diameter ratio exceeds 2.5.

C2.5.8.1.4 Louvers

Louvers serve several purposes on air coolers. By modifying the flow of air, louvers control the bundle temperature, which is perhaps the most important louver characteristic because it inhibits the process fluid from freezing. Moreover, louvers help protect the fan and/or bundle from inclement weather, such as hail and rain. Finally, louvers are effective sound dampers.

Louvers can be placed at both air cooler inlets and outlets. The louver correlation used by ***Xace*** is

$$\Delta P_{lvr} = 10^{a_1} V_{bo,st}^{b_1} \tag{C2.5-18}$$

Equation (C2.5-18) is noteworthy for multiple reasons. First, it is a dimensional equation: when it is used with the variables listed in Table C2.5-3, the velocity must be in ft/min, which yields a pressure drop in inches of water. Second, the velocity used in Equation (C2.5-18) must be corrected for standard conditions via

C2.5.8.1.4 Louvers, continued

$$V_{bo,st} = V_{bo}\left(\frac{\rho_{bo}}{0.075}\right) \tag{C2.5-19}$$

where 0.075 is the standard air density in lb/ft^3 at 1 atm and 70 °F. Equation (C2.5-19), like Equation (C2.5-18), is dimensional, and $\rho_{,bo}$ must be in lb/ft^3.

Values of the Equation (C2.5-18) variables a_1 and b_1 are compiled in Table C2.5-3, along with a representative pressure drop, remarks on the louver tested, and the source reference. The digits presented for both a_1 and b_1 greatly exceed the significant digits in order to preserve the calculation accuracy of the very sensitive power of 10 function.

C2.5.8.1.5 Steam Coils

The purpose of using steam coils is to prevent process fluid freezing. Steam coils are placed on the air intake side of the bundle and are typically disabled when freezing is not an issue. Frequently steam coils are used in conjunction with louvers and forced convection within the bundle, but combination effects are not explicitly modeled in ***Xace***.

Xace has the option of calculating an auxiliary pressure drop due to steam coils. If users wish to model this condition, they toggle *yes* for the steam coil option on the Optional panel. Otherwise, if the radio button is left on the default of *no*, this subroutine is skipped. When steam coil pressure drop is calculated, the set of equations used is [11, 12]

$$\Delta P_{sc} = \frac{2G_{\max}^2 N}{\rho_{be}\, g_c}\left(f + a_a\right) \tag{C2.5-20}$$

$$f = f_{is}\, \gamma_{rp}\, \phi_p \tag{C2.5-21}$$

$$A_{\min} = W_b\, L - \frac{W_b\, L\, D_r}{P_t} - \frac{F_\rho\, W_b\, L\, F_T\left(2F_H\right)}{P_t} \tag{C2.5-22}$$

$$G_{\max} = \frac{W}{A_{\min}} \tag{C2.5-23}$$

Table C2.5-3. Louver Pressure Drop Versus Air Velocity Data[a]

a_1	b_1	$\Delta P\|_{V=600\ ft/min}$, in. H_2O (10^2)	Remarks	Reference
-7.1825	1.9795	2.1	Original source unknown, based on bundle outlet standard conditions velocity	*Xace*
-6.0128	2.0849	60	4" – 9" 16 ga. galvanized steel or aluminum blades, 24" × 24" – 24" × 96", 4" depth, exhaust flow assumed, based on face velocity, conditions unspecified	13
-5.9001	1.8324	16	10' × 4', 1.25" vane spacing, *s*-shaped blade, exhaust flow assumed, velocity basis and conditions unspecified	14
-6.7977	2.0041	5.9	Intake: 20 ga. galvanized steel blades, 4" depth, 45°, 19 ga. ½" × ½" birdscreen, based on free area standard conditions velocity	15
-6.9099	1.9946	4.3	Exhaust: 20 ga. galvanized steel blades, 4" depth, 45°, 19 ga. ½" × ½" birdscreen, based on free area standard conditions velocity	15
-6.8262	2.0114	5.8	Intake: 12 ga. galvanized aluminum blades, 4" depth, 37.5°, 16 ga. ¾" × ¾" birdscreen, based on free area standard conditions velocity	16
-6.8291	1.9912	5.0	Exhaust: 12 ga. galvanized aluminum blades, 4" depth, 37.5°, 16 ga. ¾" × ¾" birdscreen, based on free area standard conditions velocity	16
-5.9464	1.7749	9.7	Intake: 12 ga. extruded aluminum flat blades, 4" depth, 45°, 19 ga. ½" × ½" birdscreen, based on free area velocity, conditions unspecified	17
-6.7065	1.9338	4.6	Exhaust: 12 ga. extruded aluminum flat blades, 4" depth, 45°, 19 ga. ½" × ½" birdscreen, based on free area velocity, conditions unspecified	17
-6.1554	1.7998	7.0	Intake: 12 ga. extruded aluminum stepped blades, 4" depth, 45°, 19 ga. ½" × ½" birdscreen, based on free area velocity, conditions unspecified	17
-6.8683	1.9425	3.4	Exhaust: 12 ga. extruded aluminum stepped blades, 4" depth, 45°, 19 ga. ½" × ½" birdscreen, based on free area velocity, conditions unspecified	17
-6.4861	1.9823	10	15 ga. extruded aluminum, 2" – 4" depth, exhaust flow assumed, based on free area velocity, conditions unspecified	18
-6.4251	1.8226	4.3	11 ga. extruded aluminum, 4" depth, exhaust flow assumed, based on free area velocity, conditions unspecified	18

[a] US customary units are used in this table because the corresponding equation is unit-specific

C2.5.8.1.5 Steam Coils, continued

$$\mathrm{Re} = \frac{G_{\max} D_r}{\mu_{be}} \tag{C2.5-24}$$

However, ***Xace*** is very specific in its calculation of steam coil pressure drop. It assumes that the steam coil has only one row of high-finned tubes with a 2.5-in. transverse pitch and that the width of the coil matches the width of the bundle. Furthermore, the tubes have a 1-in. root diameter, fin heights of 0.62 in., a fin density of 11 fins/in., an average fin thickness of 0.01275-in., and a fin metallurgy of aluminum. Moreover, the pressure drop row correction factor is taken as 1.041 and the pressure drop physical property correction factor used is 1.05. The isothermal friction factor was obtained from HTRI wind tunnel experiments [19].

$$f_{is} = 0.217 + \frac{1150}{\mathrm{Re}} - \frac{0.362}{\mathrm{Re}^{0.2}} \tag{C2.5-25}$$

C2.5.8.1.6 Hail Screens

Hail screens are positioned at the top of an air cooler and act as the first line of defense against structural damage, particularly hail. They are used with both forced and induced draft geometries. In the forced draft configuration, hail screens protect the bundle, whereas in induced draft, they shield the fan. Hail screen pressure drop is calculated with

$$\Delta P_{hs} = \frac{K \rho_{be} V_{fan}^2}{2} \tag{C2.5-26}$$

$$K = 0.752879 - 0.00789865 S_{nf} \; ; \; 0 < K < 1 \tag{C2.5-27}$$

C2.5.8.1.7 All Other Obstructions

Xace estimates the pressure drop resulting from any other type of flow obstruction, such as the fan motor, using

$$\Delta P_{obs} = \frac{K \rho_{be} V_{fan}^2}{2} \tag{C2.5-28}$$

with K calculated via

$$X_{opn} = 100 - A_{blk} \tag{C2.5-29}$$

C2.5.8.1.7 All Other Obstructions, continued

$$K = 2.303 - 7.73\left(10^{-3}\right)X_{opn} - 1.53\left(10^{-4}\right)X_{opn}^{2} \qquad \text{(C2.5-30)}$$

The blocked area should be based on the relevant airflow area. For example, consider a motor of 5 ft^2 cross-sectional area obstructing the intake. With a forced draft air cooler that has a fan ring intake cross-sectional area of 50 ft^2, A_{blk} would equal 10 %. An induced draft air cooler, on the other hand, with the air entering across a 100 ft^2 section of the bundle, would have an A_{blk} value of 5 %.

C2.5.8.2 Other Factors Affecting Airflow Rate

Compared to a rectangular (box) plenum, a transition plenum has no discernible effect on pressure drop. HTRI reached this conclusion after conducting tests using geometrical parameters of H/D = 0.5, P/D = 0.38.

$$P_s = f\left(q_{id}, \theta, S\right) \qquad \text{(C2.5-31)}$$

Equation (C2.5-31) describes the fan performance curve, which yields a particular flow rate at a given static pressure for a constant pitch, θ, and a constant fan speed, S. Such curves are supplied by the fan manufacturer. The fan design governs functional dependency.

The actual flow rate is calculated by multiplying the ideal flow rate from the fan curve by the appropriate correction factors:

$$q_{act} = q_{id}\, q_{nx}^{*}\, q^{**} \qquad \text{(C2.5-32)}$$

Correction factor q^{*}_{nx} represents the effect of fan height and inlet airflow blockage. Correction factor q^{**} represents the effect of the plenum depth. Subscript n designates the magnitude of the inlet airflow blockage: n = 1 indicates zero; n = 2, 3, 4, or 5 indicates inlet area blockage of 37%, 63%, 73%, or 87% respectively [4]. Subscript x designates one particular fan when three are located side by side. Table C2.5-4 gives constants for Equation (C2.5-33). The values are based on HTRI experimental data and refer to three side-by-side units: A, B, and C. Units A and C are on each end, and Unit B is in the middle. If x = A, B, or C, the correction factor belongs to any of the three fans.

Correction factors range from 0 to 1. They are given by the following equations and should be applied within the stated limits:

C2.5.8.2 Other Factors Affecting Airflow Rate, continued

$$q^*_{nx} = C_1 + C_2\left(\frac{H}{D}\right) + C_3\left(\frac{H}{D}\right)^2 \qquad \text{(C2.5-33)}$$

where

$$L_1 \le \left(\frac{H}{D}\right) \le L_2 \qquad \text{(C2.5-34)}$$

$$q^{**} = 0.947 + 0.0745\log\left(\frac{P}{D}\right) \qquad \text{(C2.5-35)}$$

$$0.1 \le \left(\frac{P}{D}\right) \le 2 \qquad \text{(C2.5-36)}$$

For $H/D > L_2$, Equation (C2.5-33) should be set equal to 1, corresponding to no correction. For $H/D > 1$, extrapolation is not recommended in the absence of data support; however, this region is usually not of industrial importance. Equation (C2.5-35) should be used within the range given by Equation (C2.5-36). For $P/D > 2$, set $q^{**} = 1$.

Table C2.5-4. Coefficients in Equations (C2.5-33) and (C2.5-34)

n	x	C_1	C_2	C_3	L_1	L_2
1	A, B, or C	1.0	0	0	0.4	4.0
2	A, B, or C	1.0	0	0	0.4	4.0
3	A, B, or C	0.781	0.0975	-0.0109	0.4	4.0
4	A or C	0.6	0.25	-0.04	0.4	3.0
4	B	0.79	0.13	-0.02	0.4	3.0
5	A or B	-0.12	1.35	-0.405	0.4	1.7
5	C	0.15	0.421	-0.0522	0.4	4.0

C2.5.8.3 Recirculation

If the fan is turned off in a particular cell while others are on in adjacent cells, downward airflow can be estimated by the following equation [4]:

$$-q_{act} = q_{id}\left[R_1 + R_2\left(\frac{H}{D}\right)^{-1} + R_3\left(\frac{H}{D}\right)^{-2}\right] \qquad \text{(C2.5-37)}$$

C2.5.8.3 Recirculation, continued

where

$$0.5 \le \left(\frac{H}{D}\right) \le 5.0 \qquad \text{(C2.5-38)}$$

Table C2.5-5 gives values for coefficients R_1, R_2, and R_3. In this case, q_{id} is the hypothetical positive airflow with the fan working. For $H/D > 5.0$, the term within brackets in Equation (C2.5-37) becomes negligibly small, signifying little recirculation. HTRI does not recommend extrapolation below $H/D = 0.5$.

Table C2.5-5. Coefficients in Equation (C2.5-37)

n	R_1	R_2	R_3
1,2,3	-0.0143	0.121	-0.029
4	0.0112	0.0632	0.0488
5	0.15	-0.0272	0.141

C2.5.9 Inert Accumulation in Air-Cooled Condensers

Nearly all condensers have a small concentration of noncondensable contaminants. In exchangers designed for one-pass multirow condensing, vapor enters the bottom row from both ends and condenses; the condensate flows out due to gravity. Noncondensables accumulate at the lowest pressure point. No matter how small the entering concentration, noncondensable accumulation continues until the pocket reaches the tube outlet. The flow pattern does not stabilize until each row except the top one has a noncondensable pocket at its outlet. Noncondensable pockets increase in size as effectiveness or number of rows increases. Figure C2.5-8 shows the simplest case.

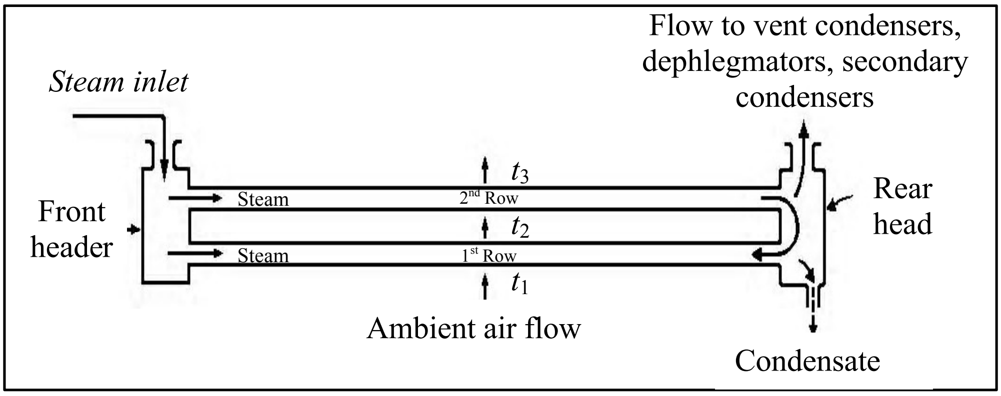

Figure C2.5-8. Trapping noncondensables

C2.5.9 Inert Accumulation in Air-Cooled Condensers, continued

Accumulation of noncondensables reduces condenser efficiency. Other adverse blanketing effects are corrosion, unstable operation, extensive tube stress, and extreme subcooling of the condensate (possibly leading to its solidification) [5].

C2.5.9.1 Effective Tube Lengths in *N*-Row Condenser with Inert Blanketing

For $j = 2, 3 \ldots N$, the $(N-1)$ equations for determining effective tube lengths for any number of rows are

$$r_1^2 = (1-E)^{2(j-1)}\left[r_1^2 + 3r_1\sum_{i=2}^{i=j} A_{i-1} + 3\left(\sum_{i=2}^{i=j} A_{i-1}\right)^2\right] \qquad \text{(C2.5-39)}$$

$$A_{j-1} = \frac{r_j - r_{j-1}}{(1-E)^{j-1}} \qquad \text{(C2.5-40)}$$

where $r_N = 1$.

Equation (C2.5-39) expands into $N-1$ numerical solution equations for the case of N rows. Figure C2.5-9 provides the plotted results for four rows.

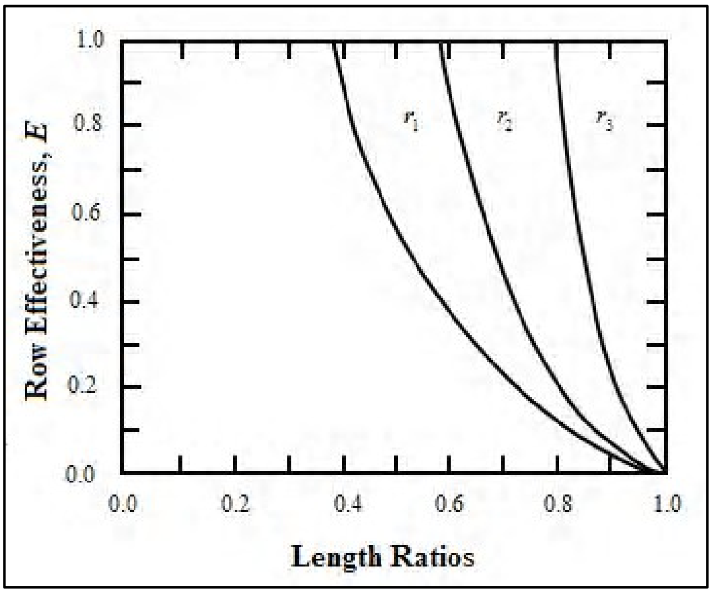

Figure C2.5-9. Effectiveness versus ratio of effective and actual tube lengths for a 4-row air-cooled condenser

C2.5.9.1 Effective Tube Lengths in *N*-Row Condenser with Inert Blanketing, continued

Equation (C2.5-41) provides the flow rate for any of the N rows (where n is a row number calculated from the air-inlet direction):

$$W_n = (w_1 L) \sum_{i=1}^{n} (r_i - r_{i-1})(1-E)^{n-1} \qquad \text{(C2.5-41)}$$

If $n = N$, then $r_N = 1$.

Equation (C2.5-42) determines the air condenser's total flow rate:

$$W = (w_1 L) \sum_{n=1}^{N} \sum_{i=1}^{n} (r_i - r_{i-1})(1-E)^{n-1} \qquad \text{(C2.5-42)}$$

C2.5.9.2 Reducing or Eliminating Inert Blanketing

Report AC-4 [6] presents ways to prevent thermal efficiency loss and eliminate adverse effects associated with noncondensable blanketing. These design modifications can be grouped into the following categories:

- variable heat transfer areas in individual rows
- excess vapor
- special tube arrangements
- single tuberow design
- flow restrictions

The specific application determines the approach required. With the exception of *variable heat transfer areas in individual rows* and *single tuberow design*, these design modifications can degrade the fluid energy on the vapor side, which might limit applicability. All modifications function properly at the design point, but may fail when a main operating variable (e.g., ambient air temperature, vapor flow, air face velocity) changes. Some approaches can work satisfactorily in a larger range of ambient temperatures if the temperature effectiveness is kept constant by controlling the face velocity.

Single tuberow design, with the most advantages, is often used for steam condensation. However, the unit usually costs more than the less effective multiple-row design.

References

1. D. D. Herrman, The application of propeller fans to air-cooled dry heat exchangers, AC-2, Heat Transfer Research, Inc., College Station, Texas (1981).

2. D. Chisholm, Air-cooled heat exchanger construction practices, AC-6, Heat Transfer Research, Inc., College Station, Texas (1988).

3. R. B. Ritter, T. P. Cham, and J. Taborek, HTRI air cooler performance evaluation – effect of fan tip clearance, ACD-2.1, Heat Transfer Research, Inc., College Station, Texas (1972).

4. D. Chisholm, Airflow problems in forced draft air-cooled heat exchangers, AC-5, Heat Transfer Research, Inc., College Station, Texas (1986).

5. G. Breber and J. W. Palen, Study of backflow and inert accumulation in air-cooled condensers, AC-3, Heat Transfer Research, Inc., College Station, Texas (1989).

6. G. Breber, Design methods which reduce or eliminate the inert blanketing in air-cooled condensers, AC-4, Heat Transfer Research, Inc., College Station, Texas (1982).

7. A. Ganguli, Fin bond resistance of air-cooled heat exchanger finned tubes, ESG-11, Heat Transfer Research, Inc., College Station, Texas (1986).

8. A. Ganguli, The effect of fin serration on the thermal hydraulics of air-cooler tubebundles, ESG-13, Heat Transfer Research, Inc., College Station, Texas (1987).

9. G. Breber, Airflow performance in induced-draft air-cooled heat exchangers, AC-7, Heat Transfer Research, Inc., College Station, TX (1990).

10. R. J. Heinsohn, *Industrial Ventilation*, 339 (1991).

11. Z. H. Yang, Development of incremental row correction factors for air-cooled heat exchangers, AC-12, Heat Transfer Research, Inc., College Station, TX (2002).

12. S. S. Tung, Experimental study of air-cooled heat exchanger performance under both natural and forced draft conditions, AC-1, Heat Transfer Research, Inc., College Station, TX (1980).

13. Empire Acoustical Systems: "Louver" Empire Acoustical Systems specifications, Internet, http://www.empireacoustical.com/Accessories/Specifications.htm (4 February 2004).

14. AmerVane VI: high velocity weather louver, Internet, http://www.aafintl.com/eprsup/aaf_intl/brochures/AmerVane%20VI%20MFAS-1-336B.pdf (4 February 2004).

15. Greenheck: FDS-402 high performance drainable blade louver, Internet, http://www.greenheck.com/pdf/louvers/FDS402.pdf (4 February 2004).

16. Greenheck: ESD-435 high performance drainable performance louver, Internet, http://www.greenheck.com/pdf/louvers/ESD435August2003.pdf (4 February 2004).

17. Pennhouses: For fresh air intake or exhaust, Internet, http://www.pennvent.com/ecatalog/GetFile.aspx?FileID=1964 (4 February 2004).

18. Shoemaker: Product and service offering, Internet, http://www.shoemakermfg.com/DLCatFull.pdf (4 February 2004).

19. A. Ganguli, W. J. Marner, C. Weierman, and J. M. Chenoweth, HTRI extended surfaces databook, B17-C, Heat Transfer Research, Inc., College Station, TX (1987).