Effectively Design Air-Cooled Heat Exchangers

This primer discusses the thermal design of ACHEs and the optimization of the thermal design, and offers guidance on selecting and designing ACHEs for various applications.

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Based on bare-tube heat-transfer area, air-cooled heat exchangers (ACHEs) cost two to three times more than water-cooled heat exchangers for the same heat duty (hardware costs only). There are two main reasons for this. First, the thermal conductivity of air is considerably lower than that of water, which results in a much lower heat-transfer coefficient. Second, since design ambient temperatures are always higher than design water temperatures, the mean temperature difference (MTD) is always lower for an ACHE, especially at relatively low process-fluid outlet temperatures.

As a result of these two factors, the heat-transfer area of an ACHE is considerably larger than that of a water-cooled heat exchanger for the same duty. In addition, the larger area requires an elaborate structural support system, which increases the cost further.

However, as all engineers know, the capital (or fixed) cost of equipment is only part of the story. What is important is the total cost — the sum of the fixed cost and the operating cost.

The operating costs for water cooling are much higher than those for air cooling. These include the costs of the initial raw water itself, makeup water, and treatment chemicals, the apportioned cost of the plant cooling tower, and the pumping cost. For ACHEs, the operating cost is just the cost of the power required to make the air flow across the tube bundles. As water becomes scarcer, the operating costs of water-cooled heat exchangers increase, thereby tilting the economics further in favor of air cooling.

This article outlines the advantages and disadvantages of ACHEs, explains and illustrates what ACHEs are, elaborates the various construction features available to accommodate different application requirements, discusses the thermal design of ACHEs and the optimization of the thermal design, and examines several special applications.

Pros and cons of ACHEs

Air-cooled heat exchangers offer several important advantages over water-cooled exchangers.

Some of these are a direct result of water not being used as the cooling medium. The high costs of using water, including the costs for the raw water, makeup water, and treatment chemicals, are eliminated. The location of the cooler, and thus the plant itself, does not depend on being near a source of water (such as a river or lake). Thermal and chemical pollution of the water source are prevented. Maintenance costs are reduced, since frequent cleaning of the waterside of the cooler (necessitated by fouling such as scaling, biofouling, sedimentation, etc.) is not needed. And, installation is simpler because water piping is eliminated.

Another advantage is that ACHEs can continue to operate, albeit at reduced capacity, by natural convection when there is a power failure.

And finally, control of the process fluid's outlet temperature (and thereby the heat duty) is easily accomplished through various methods, such as switching fans on and off, use of two-speed or variable-speed motors, use of auto-variable fans (which allow

the blade angle to be adjusted even while the fan is in motion), and so on.

Limitations. Of course, ACHEs also have some limitations.

As noted earlier, the initial capital cost of an ACHE is considerably more than that of a water-cooled unit because air has a much lower thermal conductivity and specific heat than water.

In cold climates, extensive winterization arrangements have to be incorporated to protect against freezing temperatures. This increases the initial capital costs even further.

An economical approach temperature between the outlet process fluid and the ambient air is generally in the range of 10–15°C, whereas in water-cooled exchangers this approach temperature can be as low as 3–5°C. This disadvantage is mitigated by having air cooling followed by trim cooling with water.

Because of the larger heat-transfer area, an ACHE requires a larger plot area than a water-cooled exchanger. However, this disadvantage can be overcome by locating an ACHE on a piperack so that no valuable plot area is wasted.

The low specific heat of air requires that large quantities of air be forced across the tube bundles. This is accomplished by large-diameter fan blades rotating at high speeds, which produces high noise levels.

The seasonal variation in air temperature can affect performance, and expensive control systems have to be incorporated to ensure stable operation.

ACHEs cannot be located near large obstructions, such as buildings, because air recirculation can set in and reduce efficiency.

The design of ACHEs is relatively sophisticated. Because of this, there are fewer vendors of ACHEs than there are of water-cooled shell-and-tube heat exchangers.

For cooling viscous liquids, ACHEs become even more expensive due to the extremely low tubeside heat-transfer coefficient. (Such liquids yield considerably higher heat-transfer coeffi-

cients when flowing on the outside of tubes in shell-and-tube heat exchangers, due to the much higher turbulence.) This situation can be remedied to a large extent by the use of tube inserts. However, this technology has still not become very popular.

Optimizing air and water cooling

In many applications where the process outlet temperature is relatively low, air cooling alone may not be feasible. For example, cooling a light hydrocarbon liquid to 40–45°C may not be feasible at a site where the design ambient temperature is 42°C and the design cooling water temperature is 33°C. In many such cases, a combination of air cooling followed by trim cooling with water can be adopted.

For certain other services, air cooling may not be economically viable at all. For example, at the site just described (ambient = 42°C, cooling water = 33°C), air cooling may not be viable for a naphtha stabilizer condenser with inlet and outlet temperatures of 50°C and 45°C, respectively, due to an inordinately low temperature difference. Here, water cooling alone would be recommended.

Thus, there will be services where air cooling alone is suitable, others where a combination of air and water cooling can be used, and still others where only water cooling should be employed.

The optimum temperature break-point between air and water cooling (that is, the temperature at which the process fluid leaves the ACHE and enters the water-cooled heat exchanger) has to be established by the overall economics for the specific project. It will depend on the equipment costs for the air-cooled and water-cooled heat exchangers, the total cost of using water, and the cost of power. Generally speaking, this optimum temperature is around 15–20°C more than the design ambient temperature.

The important point to note is that even for combination cooling (air plus water cooling), the ACHE will address

the major heat duty — 80% or more of the total — thereby considerably reducing the cooling water flow.

When using combination cooling, it is usually best to design the ACHE for a somewhat lower ambient temperature than would be used if there were no trim cooling and to design the trim cooler for that process fluid temperature which would be delivered by the ACHE at the maximum (or near maximum) ambient temperature. This is because the increase in the trim cooler cost will be much less than the increase in the ACHE cost if it were to handle air at the higher ambient temperature. This concept is illustrated by the following example.

Example 1. In a refinery, 71,848 kg/h of kerosene was to be cooled from 183°C to 43°C. The maximum and minimum ambient temperatures at the site were 42°C and 18°C, respectively. Cooling water was available at 33°C. The other relevant process parameters were: fouling resistance of kerosene = 0.0004 h•m²•°C/kcal, fouling resistance of water = 0.0004 h•m²•°C/kcal, allowable pressure drop of kerosene = 1.0 kg/cm², allowable pressure drop of water = 0.7 kg/cm², kerosene viscosity = 0.11 cP at 183°C and 0.9 at 43°C, and total heat duty = 5.87 MM kcal/h.

The optimum break-point temperature between air and water cooling was calculated to be 58°C. Thus, an ACHE was designed to cool the kerosene from 183°C to 58°C and a trim cooler (using water) to cool the kerosene from 58°C to 43°C. Because there was to be a trim cooler, 38°C was used as the design ambient temperature.

The ACHE was designed with one section having two tube bundles in parallel and a total bare-tube area of 327 m². Since the section had a high length-to-width ratio (12.5 m × 3.8 m), it uses three 9-ft-dia. fans.

A simulation of the ACHE's performance established that if the ambient temperature were 42°C, the kerosene outlet temperature would ˎbe 60°C. Therefore, the trim cooler was designed for this heat duty, that is, for cooling the kerosene from 60°C to 43°C. A single

shell with a heat-transfer area of 157 m²
was found to be adequate.

Now consider what would happen if
the ACHE were to be designed for an
ambient temperature of 42°C and the
trim cooler for cooling the kerosene
from 58°C to 43°C. The ACHE bare-
tube area would increase from 327 m²
to 369 m² (and the fan diameter from 9
ft to 10 ft), whereas the trim cooler
heat-transfer area would decrease from
157 m² to 137 m². Thus, it would be
more economical to design the ACHE
for an ambient temperature of 38°C
and the trim cooler for a kerosene tem-
perature equal to the outlet temperature
from the ACHE (that is, not 58°C but
60°C) when the ambient temperature is
the maximum (42°C).

Construction features

The American Petroleum Institute's
standard API 661, "Air-cooled Heat Ex-
changers for General Refinery Ser-
vices" *(1)*, outlines the minimum re-
quirements for design, materials, fabri-
cation, inspection, testing, and prepara-
tion for shipment of ACHEs. Although
this standard is intended specifically for
the petroleum refining industry, its use
is widespread in the petrochemical, fer-
tilizer, and general chemical industries
as well. In fact, it is considered to be a
standard for all ACHEs.

A discussion of construction fea-
tures should begin with a few basic
definitions:

Tube bundle — assembly of tubes,
headers, tube supports, and frames
(Figure 1).

Bay or section — the smallest inde-
pendent part of an ACHE, complete
with its tube bundles, fans, drives, mo-
tors, supporting structure, and so on
(Figure 2).

Unit — one or more bays (sections)
for an individual service, such as a
condenser or a cooler.

Bank — two or more units located
one after another on the same continu-
ous structure (Figure 3).

A bank consists of two or more
units, a unit consists of two or more
bays, and a bay consists of two or more
bundles, as illustrated in Figure 4.
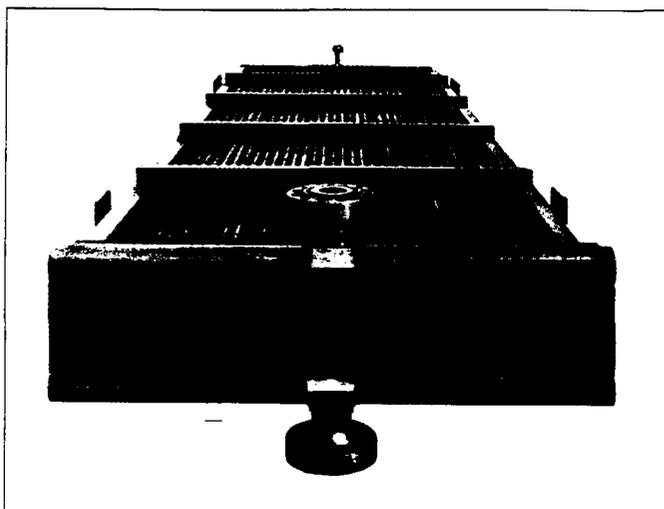
■ *Figure 1. A
tube bundle is
an assembly of
tubes, headers,
tube supports,
and frames.*
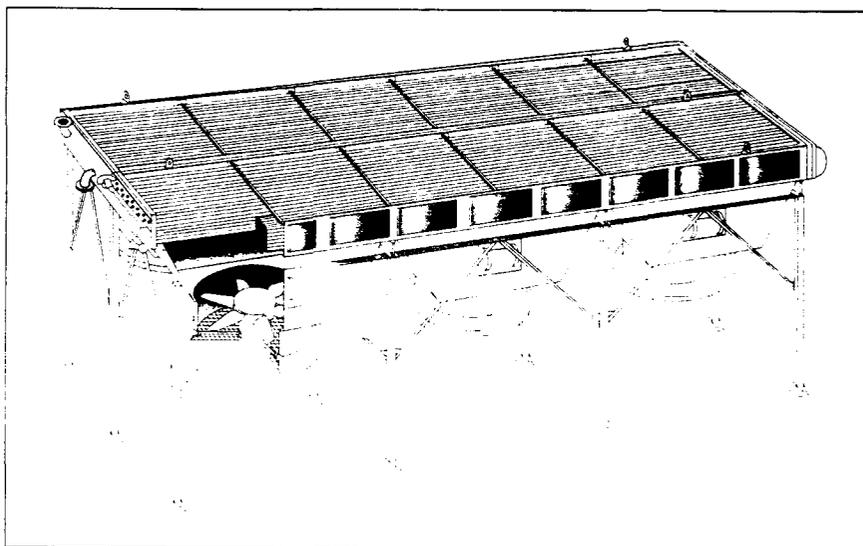
■ *Figure 2. A bay (or section) consists of two or more tube bundles.*

Forced draft — tube bundles located
on the discharge side of fans (Figure 5a).

Induced draft — tube bundles locat-
ed on the suction side of the fans (Fig-
ure 5b).

Finned tubes

The tube bundles are the heart of an
ACHE. The extremely low heat-trans-
fer coefficient of air makes the use of
extended surface tubes (on the airside)
unavoidable. The most common and
least expensive form of extended sur-
face is the finned tube.

The base tube may be made of any
commercially available material —
usually carbon steel or stainless steel.

The fins are of circular cross-section
and are normally made of aluminum,
since it has the most favorable thermal-
conductivity-to-cost ratio as well as
good cold-working properties. Because
these fins can be as thin as 0.4 mm, it is
common to pack in 433 fins per meter.

For maximum operating tempera-
tures in excess of 400°C, carbon steel
fins are used instead of aluminum.
These fins are much thicker than alu-
minum fins (usually about 0.8 mm). If
the atmospheric conditions are corro-
sive to carbon steel, the entire finned
tubes (carbon steel tubes and carbon
steel fins) are galvanized from the out-
side (tube outer surface and total fin
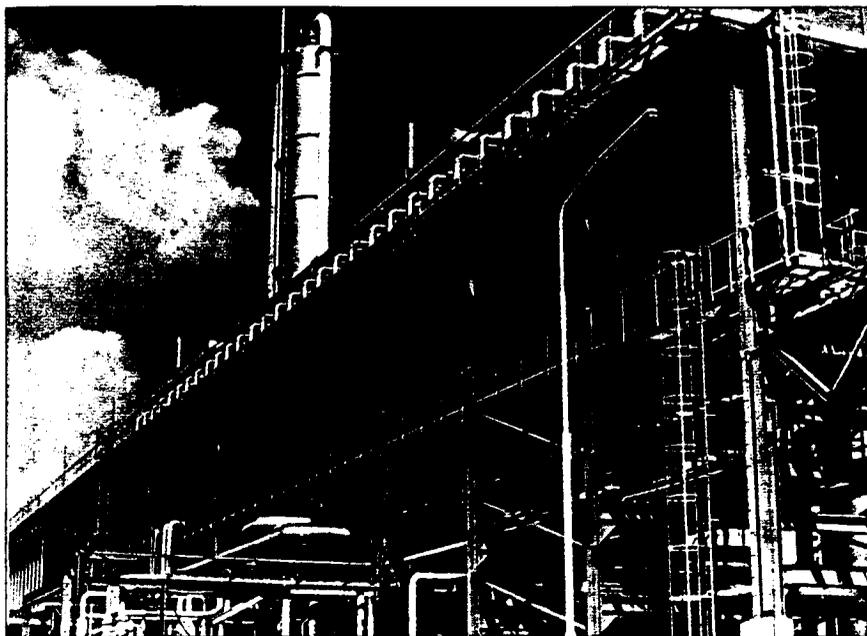
Figure 3. A bank consists of two or more units on the same structure. (Photo courtesy of ABB Lummus Heat Transfer.)

surface). However, galvanized carbon steel finned tubes cannot be used for process fluid temperatures above 300°C. The use of these tubes is not very prevalent. (See *(2)*.)

Since the coefficient of linear expansion of aluminum is about twice that of carbon steel, a gap resistance between the tube and the fin material develops. As the operating temperature increases, the difference in the coefficients of expansion increases and so does the resulting gap resistance. Thus, maximum operating temperatures, which depend on the type of bond between the tube and the fin, have been established. (These are specified for the various types of finned tubes in the following discussions.)

The standard tube O.D. is 1 in., although 1¼ in., 1½ in., and even 2 in. diameters are employed in cases where the allowable tubeside pressure drop does not permit the use of 1-in.-O.D. tubes. The standard fin heights are ⅜ in., ½ in., and ⅝ in., with the latter two being far more popular.

Single, L-footed finned tubes. This is a circular fin wrapped helically around the tube under tension (Figure 6a). Full coverage of the base tube by

the L-foot offers good protection against atmospheric corrosion.

These fins tend to become loose over time, resulting in significant deterioration in the airside performance due to the gap between the tube and the fins. Consequently, their use is limited to applications where the process-fluid inlet temperature is less than 120°C. However, the majority of ACHE applications have process inlet temperatures less than this.

Since the airside performance of these finned tubes is likely to deterio-

rate due to the loosening of the fins, they are generally not very popular. One place where they are common is in the corrosive marine atmospheres of offshore platforms, since they afford good protection against atmospheric corrosion of the base tubes and are superior to grooved fins.

Double, L-footed finned tubes. These offer better coverage of the base tube (Figure 6b). However, because they are more expensive (about 10–15%) than single L-footed finned tubes, they are preferred only in extremely corrosive atmospheres. The upper limit on process-fluid inlet temperatures for these finned tubes is 170°C.

Grooved (or embedded) or G-type finned tubes. Here the fin is embedded in the tube by first plowing a groove in the tube wall and then stretching the fin material into the groove under sufficient tension to achieve a specified bond strength (Figure 6c). These are the most commonly used finned tubes in ACHEs.

G-finned tubes require a heavier tube wall than L-footed tubes. API 661 (Clause 4.1.11.3) specifies a minimum tube wall thickness of 2.1 mm for carbon steel and low-alloy steel, and 1.65 mm for stainless steel. For embedded-fin tubes, this thickness is measured at the bottom of the groove. Hence, embedded fin tubes have to be thicker than L-footed fin tubes by the groove depth (which is usually 0.3 mm).
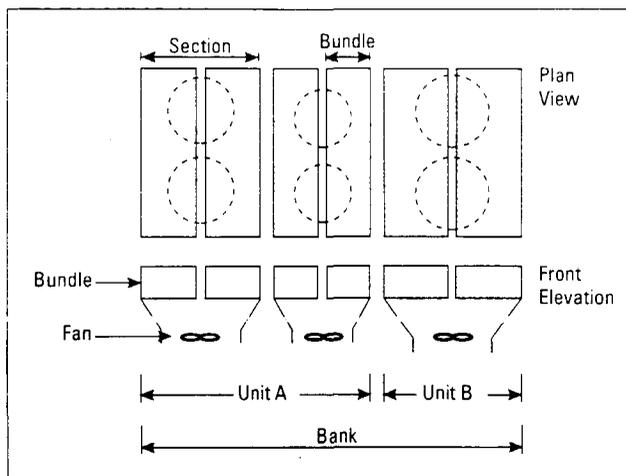
Figure 4. ACHEs consist of bundles combined into bays, bays combined into units, and units combined into banks.

■ *Figure 5. The ACHE's fans can be in a forced-draft (a) or induced-draft (b) configuration.*
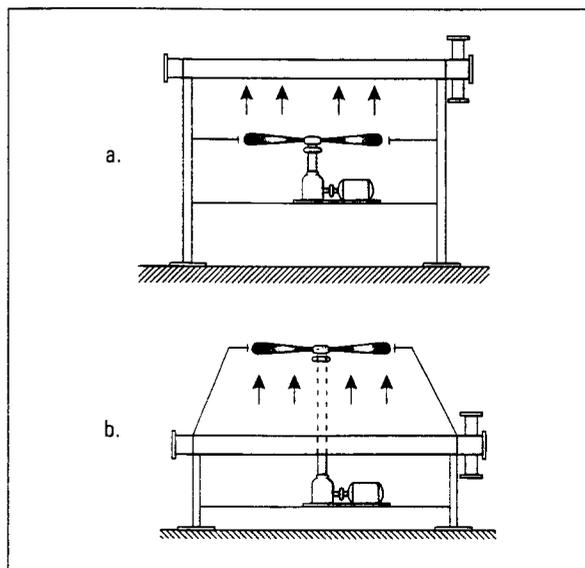
These finned tubes can withstand process-fluid inlet temperatures of up to 400°C due to the strong fin bond (the fin/bond resistance is considered negligible). They can also withstand cyclic operation without any loss of fin/tube contact. The disadvantage of G-finned tubes is that the base tube material is exposed to the atmosphere, so their use in aggressive atmospheres (such as marine) is not recommended.

Bimetallic finned tubes. Bimetallic finned tubes have G-type fins embedded in an outer tube of aluminum that is stretched over the base tube (Figure 6d). These tubes are not used very often, but they are well-suited for applications where the process fluid is at high pressure and corrosive, thereby requiring the use of an expensive alloy. In these cases, it may be cheaper to use bimetallic finned tubes with a thin-wall inner tube than to use a heavy base tube with G fins.

Extruded finned tubes. These are basically double tubes. The usual construction is an inner steel tube encased in an outer tube of aluminum (Figure 6e). The fins are extruded from the outer tube by a cold-working process. Applications are similar to those of bimetallic finned tubes.

For a detailed discussion of bond resistance and the maximum operating temperatures for various types of high-finned tubes, see *(3)*.

Headers

Headers introduce the hot fluid into the tubes and collect the cooled fluid. They carry the inlet and the outlet nozzles as well as the partition plates that create the required number of tube passes.

Headers are arranged so that movement within the side frame is possible to accommodate thermal expansion. In accordance with API 661 (Clause 4.1.6.1.2), if the temperature difference between the inlet to one pass and the outlet from the adjacent pass is greater than 111°C, split headers are used to contain the differential expansion. A split header is, literally, a header split into two (or more, in exceptional circumstances of very high tubeside fluid temperature difference) headers, one resting over the other. The upper header can slide along the top of the lower header to accommodate differential expansion between the tubes and the headers.

There are various types of header construction, each having specific advantages and disadvantages. The most common headers are as follows.

Plug headers. This is the most common construction. It consists of a rectangular, welded box with inlet nozzle(s) in the top plate and outlet nozzle(s) in the bottom plate (Figure 7a). The tubes are either welded to the tubesheet or expanded into holes in the tubesheet. The plug hole opposite each tube in the plug-sheet allows mechani-

cal cleaning of each tube and plugging in case of leakage.

This type of header is relatively cheap and can be used for pressures up to 175 bar. The disadvantage of plug headers is that for frequent cleaning of tubes in dirty services (where the fouling resistance is greater than 0.0004 h•m²•°C/kcal), removal of the large number of plugs is time-consuming and costly. Thus, plug headers are preferred for clean and moderate- to high-pressure (40–175 bar) applications.

Cover-plate headers. For dirty services, cover-plate headers (Figure 7b) are preferred because it is much easier to remove the cover plates than the numerous plugs of plug headers. For easy removal, cover-plate headers are equipped with jack-screws and lifting lugs.

At higher pressures, this header type becomes expensive as the cover-plate thickness increases. Therefore, cover-plate headers are usually not used for pressures in excess of 40 bar.

Manifold headers. For very high pressures, round manifold headers (Figure 7c) are generally used. The tubes are welded to the manifold by means of stubs.

Due to manufacturing limitations, the number of tube rows per manifold is restricted to one or two. Thus, the choice of number of tube passes is limited. For example, if a manifold has one row of tubes, the number of tube passes has to be equal to the number of rows. If a manifold has two rows of tubes, the number of tube passes has to be half the number of tube rows.

Since cleaning of the insides of the tubes can be carried out only chemically or by cutting the U-bends, this type of construction is not recommended for dirty services. However, for pressures above 175 bar, it is virtually mandatory that manifold headers be used.

Bonnet headers. In a bonnet construction (Figure 7d), a semicircular bonnet is welded or bolted to the tubesheet. This is a relatively cheap construction, but it has an inherent disadvantage in that the piping must be removed for cleaning or even for plugging a leaking tube.
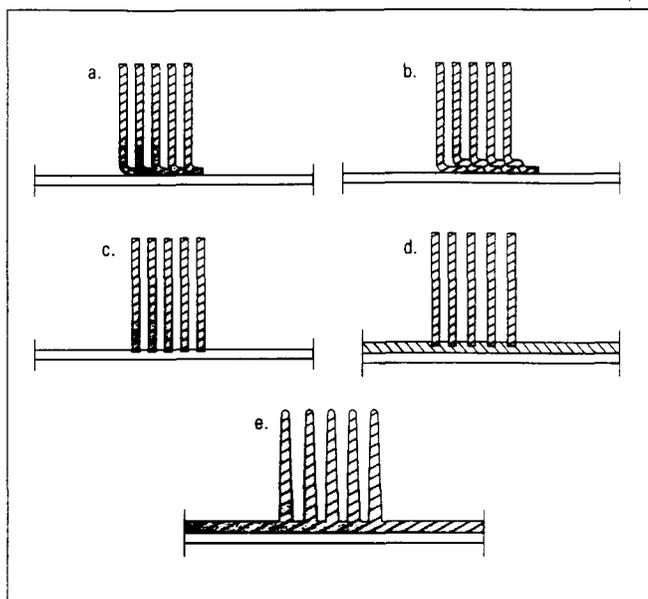
■ *Figure 6. Finned tubes are available in a variety of fin styles: L-footed (a), double L-footed (b), grooved G-type (c), bimetallic (d), and extruded (e).*
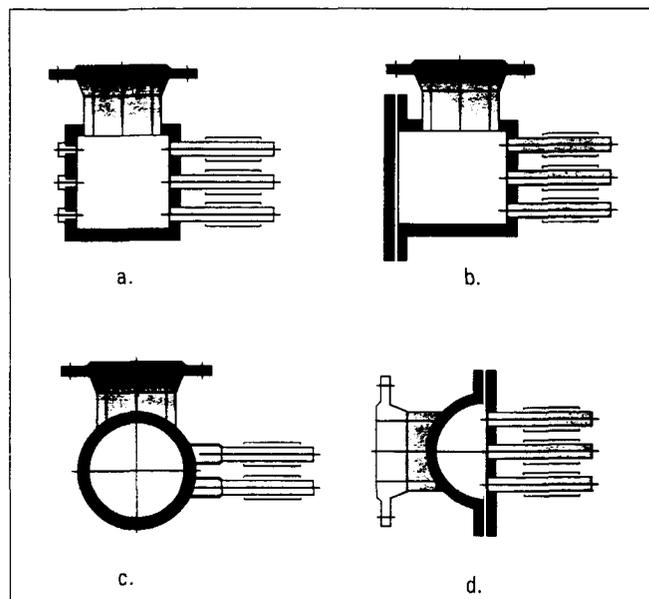
■ *Figure 7. ACHEs can use plug (a), cover-plate (b), manifold (c), or bonnet (d) headers.*

Other tube bundle features

Tube supports. The finned tubes are supported by special aluminum support boxes or by zinc collars cast on the tubes themselves.

Side frames. The ACHE's side-frame serves two purposes. First, it supports the headers and the tubes and makes the tube bundle a rigid, self-contained assembly that can be transported and erected conveniently. Second, it serves as a seal and prevents the bypassing of air.

Tube-to-tubesheet joint. At low to medium pressures (less than 70 bar), the tubes are expanded into double grooves in the tubesheet. At high pressures (greater than 70 bar), however, the tubes are strength-welded to the tubesheet. (Strength-welding produces a proper fillet-welded joint with a much higher strength than a seal-welded joint, which has no fillet.)

Fans and drives

Axial fans employed in ACHEs displace very large volumes of air against a low static pressure (typically 0.6–0.8 in. w.g.). These fans have characteristic performance curves that are specific to each fan manufacturer. The designer should have access to fan curves that provide information regarding volume (or mass) of air, static pressure, absorbed power, and noise. Some fan manufacturers even furnish computer software to aid the designer in proper fan selection.

Fan diameter usually varies between 6 ft and 18 ft, although fans with smaller and larger diameters are employed in special circumstances.

A fan consists of two basic components — the hub and the blades.

The hub is mounted on the fan shaft and the blades are mounted on the hub. The hub may be made of cast iron, cast aluminum, or fabricated steel. Manufacturers usually conduct static and dynamic balancing of the hub in the shop.

Two types of hubs are available. With a manually adjustable hub, the blade angle can be altered only when the fan is stationary. An auto-variable hub includes a device (usually a pneumatic controller) that can alter the blade angle even while the fan is in motion, in order to control air flow. Control is usually effected by means of a signal from a temperature indicator/controller (TIC) responding to the outlet temperature of the process fluid. (See *(4)*.)

Blades can be of either metal (usually aluminum) or fiberglass-reinforced plastic (FRP). Plastic blades are suitable only for temperatures of up to 70°C.

Fan performance (air flow rate and static pressure) is determined by the number of blades, blade tip speed, blade angle, and blade width.

The effect of a change in tip speed on a fan's performance is dramatic. The volume of air flow varies directly with the tip speed, the pressure varies as the square of tip speed, and the power consumption varies as the cube of tip speed. Tip speed is normally limited to 61 m/s, as noise becomes excessive at higher speeds.

Increasing the number of blades increases the fan's ability to work under pressure. Thus, one could use a 6-blade fan operating at a lower tip speed to deliver the same volume of air as a 4-blade fan. However, this can be carried too far — as the number of blades increases beyond six, multi-blade interference may actually reduce the efficiency of the fan, since each blade works in the disturbed wake of the preceding blade. Therefore, the number of blades has to be selected carefully by the fan vendor.
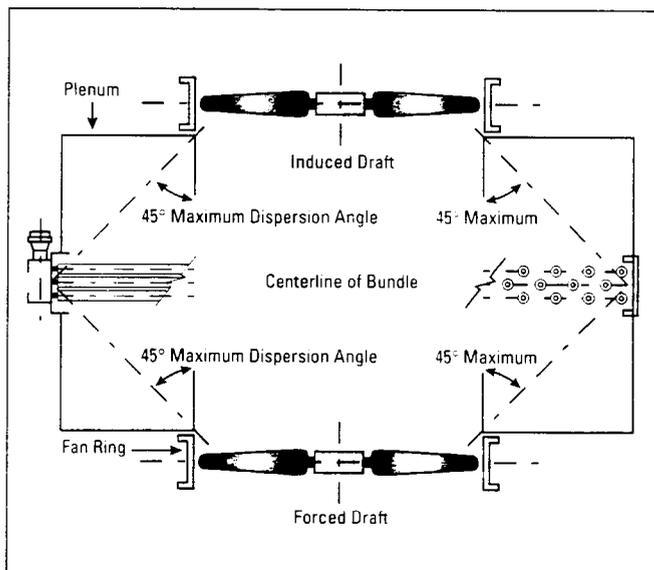
■ *Figure 8. The dispersion angle of the fan should be less than 45°.*
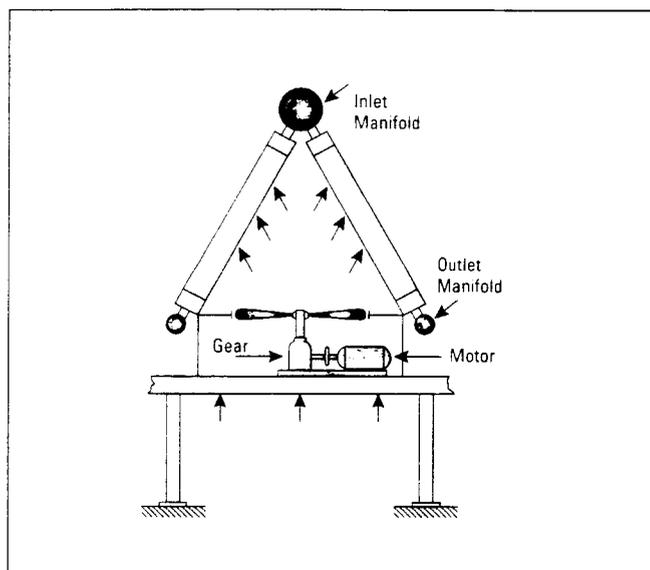
■ *Figure 9. An A-frame configuration is well-suited to condensing services.*

All the blades of a fan should be set at the same angle for smooth operation. Usually, the blade angle is set between 12° and 27°. This is because performance deteriorates at lower angles and becomes unstable at higher angles.

A fan with a wider blade can be operated at a lower tip speed to achieve the same performance. Consequently, fans with wider blades operate less noisily. This feature is exploited by fan vendors who offer special low-noise fans.

API 661 (Clause 4.2.3) stipulates the following for fans and fan hubs for ACHEs:

1. There should be at least two fans along the tube length. However, a single-fan design can be used in exceptional circumstances (such as very small units) if agreed to by the purchaser and the vendor. This two-fan requirement is apparently based on considerations of reliability — should one fan stop functioning, the other will be running and the unit will continue to operate, albeit at a somewhat lower heat duty. Furthermore, at lower loads and at cooler ambient temperatures, one fan may be stopped for better control of the process outlet temperature and to conserve power. When auto-variable fans are used, both fans need not be auto-variable — one auto-vari-

able fan and one manually adjustable fan offer the necessary control.

2. Fans should be of the axial-flow type and each fan should occupy at least 40% of the tube-bundle face area served by it.

3. The dispersion angle of a fan (Figure 8) should not exceed 45° at the centerline of the tube bundle.

4. The radial clearance between the fan ring and the fan tip should not exceed 0.5% of the fan diameter or 19 mm, whichever is less. Fan stalling may occur at larger clearances.

5. Fan tip speed should not exceed 61 m/s unless approved by the purchaser, and in no case should it exceed 81 m/s.

Some additional recommended design guidelines are as follows.

The minimum distance between the plane of the fan and the tube bundle (that is, the plenum height) should be one-half the fan diameter for forced-draft units and one-third the fan diameter for induced-draft units. These requirements are for maintaining favorable aerodynamics and thereby superior performance of fans.

For both forced-draft and induced-draft designs, the height of the fan ring should be at least one-sixth the fan diameter.

Air seals should be provided between tube bundles and between tube bundles and the plenum chamber in order to minimize air bypassing. Any gap wider than 1 cm should be considered excessive.

The fan drive provides the power required by the fans to move air across tube bundles. Fans can be driven by an electric motor, steam turbine, gas or gasoline engine, or hydraulic motor. The electric motor is overwhelmingly the most popular. Polyphase, squirrel-cage totally enclosed fan-cooled induction motors are usually used. The requirements for steam turbine drivers are specified in API 611 (5).

The power is transmitted from the motor (or turbine) to the fans through a direct drive, V-belt drive, high-torque drive (HTD), or gear drive. API 661 (Clauses 4.2.8-2.10, -2.11, -3.1, -3.3) outlines the maximum fan power ratings for each type of drive. A direct drive, where the fan shaft is directly connected to the driver, is usually used with fan diameters of 5 ft (1.53 m) or less and drives of 5 hp (3.73 kW) or less. A V-belt drive is used when the speed rating (rpm) of the fan is less than the speed rating of the driver. V-belt drives may be used with motor drives rated 30 hp (22 kW) or less. An

HTD may be used with motor drives rated 50 hp (37 kW) or less. For electric motors rated above 50 hp (37 kW). right-angle gear drives must be used. All steam turbine drivers must employ right-angle gear drives

Unlike flat and V-belts. HTD belts do not rely on friction for their pulling power. Rather, HTD belts utilize a new tooth design that substantially improves stress distribution and higher overall loading.

HTD belts do not stretch due to wear, are corrosion-resistant, and operate at reduced noise levels. The belts are capable of transmitting higher torque at lower speed, thus improving the power capacity of toothed belts.

Maintenance is simple. No adjustments are required due to stretch or wear. HTD belts are ideal where maintenance is difficult or where downtime could prove to be extremely expensive.

The plenum chamber is where the air delivered by a fan is distributed to (forced-draft) or collected from (induced-draft) the tube bundle. It consists of a rectangular box or a conical/rectangular transition piece. For forced-draft units. the plenum chamber can be square (or rectangular) or conical. whereas for induced-draft units. it is invariably conical. A partition is provided between fans. and the gap between the tube bundle and the partition plate should not exceed 20 mm.

For forced-draft units. the plenum chamber has a conical inlet at the bottom to reduce inlet losses. When low-noise fans are employed. this conical inlet is replaced by a bell-shaped mouth.

Configuration of ACHEs

Horizontal configuration. ACHEs are usually configured in the horizontal position (Figure 5) because this makes maintenance easier.

A-frame configuration. This design (Figure 9) is employed almost exclusively in power plants for condensing turbine exhaust steam. The tube bundles are mounted on a triangular frame with fans located below. The in-

clination from the horizontal is usually between 45° and 60°.

The A-frame configuration occupies 30–40% less plot area than a horizontal configuration. Additionally, though no less importantly, the A-frame is ideal for condensing, as it facilitates condensate drainage. The common header at the top of the unit allows uniform steam distribution with minimum pressure loss, which is important for the efficient operation of vacuum steam condensers. The A-frame configuration is, in fact, the basis of several patented freeze-proof designs.

Vertical configuration. Vertical orientation (Figure 10) is generally employed for packaged systems, such as compressors with their intercoolers. These units are used where floor space is at a premium.

A drawback is that they are much more prone to performance deterioration due to cross-winds. Furthermore, multipass designs are not feasible for condensing services.

Natural vs. mechanical draft

Natural draft involves no fans — air flow is by natural convection due to the stack effect across the tube bundle. An external stack is sometimes installed to increase the draft and thereby the cooling. The principal application of natural draft is in dry cooling towers in power plants, where the large chimney of the cooling tower establishes an appreciable draft.

Most ACHEs are of the mechanical-draft design. Vast amounts of air are moved across finned-tube bundles by axial fans driven by electric motors. There are two main types of mechanical draft — forced draft and induced draft. In forced draft (Figure 5a), fans mounted below the tube bundles blow air across the finned tubes. In induced draft (Figure 5b). fans above the tube bundles suck air across the finned tubes. Each type has advantages and disadvantages, and therefore preferred applications.

Forced-draft ACHEs. Forced-draft ACHEs have several advantages. Because both fans and motors or drive transmissions are located below the tube bundles, accessibility for maintenance is much better. Because the fans are located below the tube bundles and handle the colder incoming air, the air pressure drop, and therefore the fan power consumption, is somewhat lower. Fan blade life is longer, since exposure is to cold inlet air. Furthermore, even for relatively high air outlet temperatures (greater than 70°C), fiber-reinforced plastic blades can be used.

There are four principal disadvantages of forced draft. First, distribution of air across the tube bundles is poorer, since the air leaves the tube bundles at a much lower velocity.

Second, hot air recirculation is more likely to occur, due to the lower discharge velocity and the absence of a stack, resulting in a higher air inlet
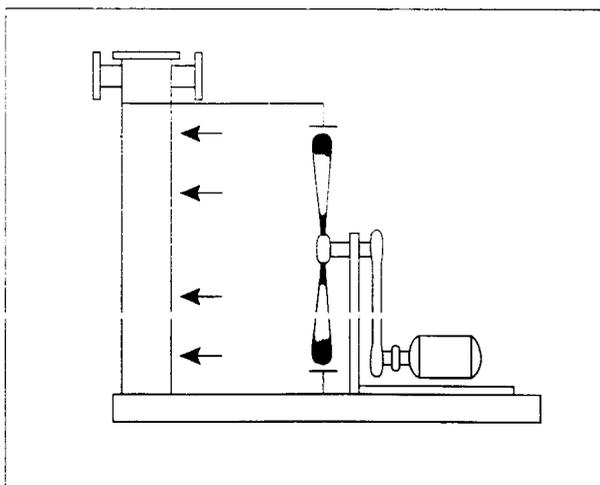
temperature and consequently a lower MTD. In low-MTD applications, the deterioration in performance could be significant. Consequently, forced draft is not preferred where the cold-end temperature approach (the difference in temperature between the process outlet and the inlet air) is less than 5–8°C.

Third, the ACHE is exposed to the elements (sun, rain, hail, and snow) unless louvers or roofs are provided at the top of the tube bundles. This results in poorer stability and process control.

Finally, due to a very small stack effect, natural draft capability in the event of fan failure is rather low.

Induced-draft ACHEs. The principal advantages of induced draft are: better air distribution across the tube bundles; lower probability of hot air recirculation (the air velocity at the discharge is usually more than twice that at the entrance); greater ability to operate following fan failure (due to the much higher stack effect); better process control and protection from the effects of rain, snow, hail, or sun; and no possibility of damage to fans or drives due to leaking products (if corrosive).

Nomenclature

A	= heat-transfer area
D	= fan diameter in m
htc_a	= airside heat-transfer coefficient
htc_t	= tubeside heat-transfer coefficient
HP	= absorbed shaft horsepower
k	= constant established by performance tests or furnished by the fan manufacturer for calculating the overall sound power level
MTD	= mean temperature difference
PWL	= sound power level in dB (reference level = 10^{-12} watts)
Q	= heat-transfer duty
$r_{f,t}$	= tubeside fouling resistance
$r_{f,a}$	= airside fouling resistance
r_w	= wall resistance
SPL	= sound pressure level in dB (reference level = 0.0002 microbars)
TS	= tip speed in m/s
U	= overall heat-transfer coefficient

However, induced draft does have several disadvantages. The air pressure drop and power consumption are higher because the air being handled is hotter and lighter. In order to prevent damage to fan blades, V-belts, bearings, and other mechanical components, the exit air temperature has to be limited to about 90°C (70°C for FRP blades). The fans, being located above the plenum chamber, are less accessible for maintenance. Further, maintenance work may have to be carried out in the hot air caused by natural convection. Since the motor is usually located below the tube bundle, a long shaft is required to transmit the power to the fan located above the bundle. This represents another potential problem area.

Thermal design

An ACHE, like any other heat exchanger, must satisfy the following basic equation:

$$A = Q/(U)(MTD) \qquad (1)$$

The overall heat-transfer coefficient, U, is determined as follows:

$$1/U = 1/htc_a + 1/htc_t + r_{f,t} + r_{f,a} + r_w \qquad (2)$$

Since the airside heat-transfer coefficient is generally much lower than the tubeside heat-transfer coefficient, it is necessary to use finned tubes to make the airside heat-transfer coefficient more compatible with the tubeside heat-transfer coefficient.

Tubeside calculations. The tubeside heat-transfer coefficient and pressure drop calculations are performed just as they are in shell-and-tube heat exchangers, whether for single-phase cooling, condensing, or a combination of the two. This has been covered extensively in the literature (such as (6, 7)) and is rather common knowledge among chemical engineers, so it will not be discussed here.

Increased tubeside pressure drop. It sometimes happens that the allowable tubeside pressure drop is specified very low. This can penalize the ACHE design considerably by requiring an inordinately high heat-transfer surface area.

For gases and condensers, the allowable pressure drop is generally between 0.05 and 0.2 kg/cm², depending upon the operating pressure — the lower the operating pressure, the lower the allowable pressure drop. For liquids, the allowable pressure drop is generally 0.5 to 0.7 kg/cm². However, if the liquid viscosity is high, a higher pressure drop is warranted for an optimum design.

Example 2. A vacuum-column bottoms cooler was to be designed for the hydrocracker unit of an oil refinery. The principal process parameters were: heat duty = 5.764×10^6 kcal/h, flow rate = 93,278 kg/h, inlet temperature = 180°C, outlet temperature = 70°C, allowable pressure drop = 0.5 kg/cm², inlet viscosity = 4.2 cP, outlet viscosity = 15.0 cP, and fouling resistance = 0.0006 h•m²•°C/kcal. Carbon steel tubes of 25 mm O.D. and 2.5 mm thickness were to be used. Since the ACHE would be located on a 12-m-wide piperack, the tube length was to be 12.5 m. The design ambient temperature was 42°C.

Since the liquid viscosity was rather high, the allowable pressured drop of 0.5 kg/cm² was rather low. However, it was used to prepare a preliminary thermal design, as shown in Table 1.

Due to the low allowable tubeside pressure drop, the tubeside heat-transfer coefficient was only 68.6 kcal/h•m²• C, which represented 86.6% of the total heat-transfer resistance. In order to produce a more-economical design, a higher tubeside pressure drop of 1.6 kg/cm² was permitted. The revised design is also outlined in Table 1. The overdesign was 8.85%, the same as in the original design.

Note that in the revised design, the tubeside heat-transfer coefficient was 97.3 kcal/h•m²• C (compared to 68.8 kcal/h•m²•°C for the original design), and the overall heat-transfer coefficient was 80.07 kcal/h•m²•°C (instead of 59.72 kcal/h•m²•°C). As a result, the number of sections could be reduced from three to two and the overall bare-tube heat-transfer area from 2,027 m² to 1,536 m². The net result was a con-

siderable reduction in the cost and the plot area of the unit. Although the power consumption of each fan in the revised design was more (18.9 kW vs. 15.5 kW), the total power consumption of 75.6 kW was less than the 93.0 kW of the original design because there were fewer fans. The only negative effect of the revised design was that the design pressure of the vacuum-column bottoms circuit increased marginally, resulting in a very minor increase in the cost of the other heat exchangers in the circuit upstream of the ACHE.

Airside calculations. The airside heat-transfer coefficient and pressure drop calculations are rather complex because they involve extended surface. For a detailed discussion on this subject, see *(8–10)*.

However, the designer should keep in mind that whereas the airside heat-transfer coefficient varies to the 0.5 power of air mass velocity, the pressure drop varies to the 1.75 power of the same.

Mean temperature difference. The MTD calculations for ACHEs are somewhat different from those of shell-and-tube heat exchangers, since ACHEs employ pure crossflow. Thus, the MTD curves furnished in the Tubular Exchanger Manufacturers Association (TEMA) standards *(11)* are not applicable to ACHEs. One source of information on cross-flow MTD determination is *(12)*.

Interestingly, as the number of tube passes increases in an ACHE, the MTD also increases. For four or more tube passes, the MTD becomes equal to the true countercurrent MTD determined from the four terminal temperatures.

In water-cooled heat exchangers, the cooling water outlet temperature is usually limited to 43–45°C, based upon considerations of scaling by reverse-solubility salts. In the case of ACHEs, however, there is no such limitation. Consequently, the air flow rate and its outlet temperature can vary to a greater extent.

When the air mass-flow rate is lowered, the air outlet temperature increases, thereby reducing the MTD.

Table 1. Superior design with increased tubeside pressure drop (Example 2).

	Original Design, Lower Pressure Drop	Revised Design, Increased Pressure Drop
Number of sections	3	2
Number of bundles per section	2	2
Number of tube rows	8	8
Number of tubes per row	44	50
Number of tube passes	8	10*
Tubeside heat-transfer coefficient, kcal/h•m²•°C	68.8	97.3
Tubeside pressure drop, kg/cm²	0.5	1.54
Overall heat-transfer coefficient, kcal/h•m²•°C	59.72	80.07
Total air flow, kg/h	2,090,000	1,600,000
Mean temperature difference, °C	68.19	67.14
Total bare-tube area, m²	2,027	1,536
Power per fan, kW	15.5	18.9
Total power, kW	93.0	75.6
Degree of overdesign, %	8.85	8.85

*In the revised design, two passes in each of the two upper rows and one pass in each of the six remaining rows.

Furthermore, the airside heat-transfer coefficient is reduced due to the lower mass velocity, thereby reducing the overall heat-transfer coefficient. Both these effects increase the required heat-transfer area. The gain is in the power consumption, as both the reduced flow rate and the consequently lower pressure drop result in a lower power requirement.

The optimum thermal design will be the one that best balances these opposing tendencies. This balance will depend upon the extent to which the airside heat-transfer coefficient is controlling. Optimization of thermal design is discussed in detail in the next section.

Design ambient temperature. The selection of an appropriate design ambient temperature is of the utmost importance. Without realizing the consequences, many customers specify the highest temperature as the design air temperature. This is an extremely conservative practice and will result in an unnecessarily high cost for the ACHE.

For example, if a plant site has a summer peak temperature of 45°C and a winter low temperature of 2°C, the design air temperature should not be 45°C, but rather somewhat lower, say 42°C. It is not prudent to penalize the ACHE design by basing it on a peak temperature that may occur only for a few hours during the entire year. The cost difference between the two cases may be appreciable, especially if the MTD is low.

The usual practice is to select that temperature which is not exceeded during 2–3% of the total yearly hours of operation. Thus, a temperature variation chart at sufficiently short intervals throughout the year is needed for a proper estimate of the design ambient temperature. Such data are available from the meteorological office.

Example 3. An ACHE was to be designed for condensing the overhead from a naphtha splitter. The principal process parameters were: flow rate = 95,628 kg/h, operating pressure = 2.55 kg/cm² abs., allowable pressure drop =

Table 2. Effect of design air temperature (Example 3).

Design Ambient Temperature, °C	Mean Temperature Difference, °C	Degree of Overdesign, %
40	17.53	12.92
41	16.51	6.79
42	15.4	Nil
43	14.37	−6.31
44	13.31	−13.7

0.2 kg/cm², inlet temperature = 70.5°C, outlet temperature = 55°C, and fouling resistance = 0.0003 h•m²•°C/kcal.

Using a design ambient temperature of 42°C, the following thermal design was prepared: number of sections = 4, number of bundles per section = 2, tube dimensions = 32 mm O.D. × 2.5 mm thick × 10,500 mm long, fin height = 64 mm, fin density = 433 fins/m, total bare-tube area = 1,248 m².

Varying the design ambient temperature from 40°C to 44°C resulted in a sharp reduction in the MTD, with a consequent increase in the heat-transfer area. Table 2 presents the degree of overdesign at various ambient temperatures, indicating approximately how much more (or less) area would be required.

Noise. Noise pollution often represents a constraint in the design of ACHEs. Fans and drives have to be selected to comply with local noise regulations.

The sound power level, the total sound emitted by a system to the environment, is an absolute quantity. The sound pressure level is a relative quantity, and is the measured sound pressure related to a fixed value, largely depending upon the distance from the source of the noise.

The sound power level emitted by ACHEs to the environment is produced almost entirely by the fans and drives, with the fan drive often accounting for as much as half of the total. Typical sound power levels are 60–80 dB$_A$ for electric motors, 60–65 dB$_A$ for V-belt drives, and 70–100 dB$_A$ for gear boxes.

The major part of the noise produced from axial fans is by vortex

shedding at the trailing edge of the blades. The noise power produced by the fans will vary approximately to the third power of the fan tip speed. Fan manufacturers generally tend to limit the tip speed to about 60 m/s. Special low-noise fans are also designed for quieter operation at a given horsepower — these employ more blades or wider-chord blades (or both).

The overall sound power level (*PWL*) of an ACHE installation can be expressed as:

$$PWL = k + 30 \log TS$$
$$+ 10 \log HP \qquad (3)$$

For a forced-draft installation, the maximum sound pressure level (*SPL*) at a distance of 1 m from the fan is given by:

$$SPL = 46 + 30 \log TS$$
$$+ 10 \log HP - 20 \log D \qquad (4)$$

An induced-draft installation will generate an *SPL* that is 3 dB less than the *SPL* for a forced-draft system as determined by Eq. 4.

From the above, it can be seen that if the tip speed of a fan is reduced from 60 m/s to 40 m/s, the *SPL* will drop by 5.3 dB.

When two sounds of equal intensity are added together, the resulting sound level increases by 10 log 2, or 3 dB. Therefore, when the number of fans is doubled from one to two, the sound level increases by only 3 dB. The same effect results when the number of fans is doubled from two to four, four to eight, and so on.

The distance between the observer and the fan also has an important bear-

ing. The sound pressure level varies as 20 log (d_2/d_1), where (d_2/d_1) is the distance ratio. This translates into a reduction of 6 dB for a doubling of distance from the fan. Further discussion of noise generated by ACHEs can be found in (*13, 14*).

Optimization of thermal design

Optimization of thermal design is much more challenging, and rewarding, for ACHEs than for shell-and-tube heat exchangers. In the latter, the cooling water flow rate is essentially fixed by its inlet temperature and maximum outlet temperature, usually a temperature difference of 10–12°C. However, there is much more flexibility in the selection of an ACHE's air flow rate, which must be optimized carefully.

In addition to the air flow rate, the other variables that have to be optimized in terms of lowest total cost (fixed cost plus operating cost) are:

- tube length;
- tube O.D.;
- fin height;
- fin spacing;
- number of tube rows;
- fan power consumption;
- tube pitch; and
- number of tube passes.

Another complication is that fan size is linked to tube length and section width, thereby making the total optimization rather difficult. The use of a sophisticated computer program thus becomes essential for this task. [For information on such software, see *The 1997 CEP Software Directory*, published in January. — Editor]

Let us now consider the individual variables in more detail.

Tube length

The number of sections (or bays) should be minimized (as should the number of shells in series or parallel in shell-and-tube heat exchangers). Usually, there are two tube bundles in a section, although there can be more for very large units. Bundle width is usually limited to 3.2–3.5 m for convenience of transportation. Thus, a section hav-

ing the usual two tube bundles is 6.4–7 m wide.

A 12-ft (3.657 m), 13-ft (3.962 m), or 14-ft (4.267 m) diameter fan fits in this width quite comfortably. Since there must be at least two fans along the tube length as noted earlier, a tube length of 8–10 m will satisfy the minimum 40% fan coverage criterion specified in API 661.

Sometimes, an ACHE section may be highly rectangular (that is, the length-to-width ratio is 2.5 or more). This may arise, for example, in the case of a large plant with a wide piperack (and thus a long tube length) but a small number of low-duty ACHEs. In such cases, three fans may be used along the tube length. Since the cost for such designs is higher, a 3-fan-per-section design should be a last resort.

Since ACHEs are usually located on top of a piperack and across it, the tube length is determined by the piperack width. The tube length is usually 0.5 m greater than the piperack width for convenience of locating the plenum chambers.

Sometimes, ACHEs are grade-mounted. In such cases, the above limitation will not apply and tube length can be better optimized.

It is evident that the number of tube bundles will have to be an even number, as two of them are combined to form one section.

Tube O.D.

Just as in shell-and-tube heat exchangers, the smaller the tube diameter, the less expensive the ACHE will be. However, the smaller the tube diameter, the more difficult it will be to clean the tubes.

The smallest tube diameter recommended by API 661 (Clause 4.1.11.1) is 1 in. or 25 mm O.D., and most ACHEs are constructed with tubes of this size. However, in pressure-drop-limiting cases, a larger tube diameter may result in a better design. For example, a particular design with 25-mm-O.D. tubes and two tube passes may have a pressure drop exceeding the permitted value. If the number of

tube passes is reduced to one, the pressure drop will decrease to approximately one-seventh the earlier value and will be acceptable. However, the tubeside heat-transfer coefficient will decrease appreciably, thus necessitating a larger heat-transfer area. In this situation, 31-mm-O.D. tubes and two tube passes yield a much more economical design.

Example 4. An overhead condenser on a crude column was to be designed for a refinery with the following process parameters: heat duty = 19.584 × 10⁶ kcal/h, flow rate = 139,800 kg/h, inlet vapor weight fraction = 1.0, outlet vapor weight fraction = 0.04, operating pressure = 2.3 kg/cm² abs., inlet temperature = 123°C, outlet temperature = 65°C, allowable pressure drop = 0.25 kg/cm², fouling resistance = 0.0002 h·m²·°C/kcal, and design ambient tem-

perature = 38.0°C. Tubes were to be carbon steel and 12,500 mm long.

A design was prepared with 25-mm-O.D., 2.5-mm-thick tubes, with two tube bundles in each of three sections; the other construction details are listed in Table 3. With two tube passes, the tubeside pressure drop of 0.573 kg/cm² far exceeded the allowable limit of 0.25 kg/cm². Due to the rather high tubeside heat-transfer coefficient, a low airside heat-transfer coefficient could be tolerated and a low airside velocity could be maintained, leading to an unusually low power consumption. However, the design was not acceptable because of the high tubeside pressure drop.

A single-pass design was prepared next. Since the tubeside heat-transfer coefficient was significantly lower, the heat-transfer area required was much

Table 3. Details of crude column overhead condenser design with 25-mm and 32-mm O.D. tubes (Example 4).

	Design #1 25-mm O.D., 2 Passes	Design #2 25-mm O.D., 1 Pass	Design #3 32-mm O.D., 2 Passes
Number of sections	3	3	3
Number of bundles per section	2	2	2
Number of tube rows	5	6	5
Number of tubes per row	48	46	38
Fin height, mm	16	16	16
Fin density, fins/m	433	433	433
Transverse pitch, mm	67	67	73
Bare-tube area, m²	1,384	1,591	1,401
Total heat-transfer area (bare-tube + fin), m²	33,050	37,988	30,691
Number of fans per section	2	2	2
Fan diameter, m	4.572	4.572	4.267
Tubeside pressure drop, kg/cm²	0.573	0.072	0.214
Tubeside heat-transfer coefficient, kcal/h·m²·°C	2,112.8	1,027.2	1,381.5
Air flow rate, kg/h	2,130,000	2,750,000	2,420,000
Airside pressure drop, mm H₂0	5.8	10.97	11.95
Airside heat-transfer coefficient (based on bare-tube area), kcal/h·m²·°C	713.3	826.04	747.32
Overall heat-transfer coefficient, kcal/h·m²·°C	433.4	364.8	400.9
Power per fan, kW	9.72	23.12	21.87
Degree of overdesign, %	3.78	3.61	3.61

larger. The number of tube rows increased from five to six. Note that the penalty for the single-pass design (as will be discussed later) has not been included, so the actual heat-transfer area will be even larger.

Since the design with 25-mm O.D. tubes and one pass was uneconomical, the next design considered used 32-mm O.D., 2.5-mm thick tubes, and two tube passes. This design was much more economical, as the heat-transfer area was appreciably less.

Fin height

The usual fin heights are ⅜ in., ½ in., and ⅝ in. The selection of fin height depends on the relative values of the airside and the tubeside heat-transfer coefficients. Where the airside heat-transfer coefficient is controlling (that is, it is the major resistance to heat transfer), a larger fin height (15.875 mm) will usually result in a better design. If, however, the tubeside heat-transfer coefficient is controlling, it will be prudent to use a smaller fin height (12.7 mm).

A higher fin height leads to a greater efficiency in converting pressure drop to heat transfer on the airside. However, with higher fins, fewer tubes can be accommodated per row for the same bundle width.

In cases where the airside heat-transfer coefficient is controlling, a higher heat-transfer coefficient achieved by use of higher fins results in a significant increase in the overall heat-transfer coefficient, thereby reducing the heat-transfer area and thus the number of tubes per row. Hence, it usually becomes more economical to use higher fins for services where the airside heat-transfer coefficient is controlling, such as in steam condensers and water coolers.

For gas coolers and viscous liquid hydrocarbon coolers, however, the tubeside heat-transfer coefficient is controlling, so the use of shorter fins is economically advantageous.

This reasoning is illustrated by the following two examples, one in which the airside heat-transfer coefficient is controlling and another in which the

Table 4. Effect of varying fin height in an application where the airside heat-transfer coefficient controls (Example 5).

	Fins: ½-in. high	Fins: ⅝-in. high
Number of sections	2	2
Number of bundles per section	2	2
Number of tube rows	4	4
Number of tubes per row	48	Alternately 42 and 43
Transverse pitch, mm	60	67
Fin density, fins/m	433	433
Airside heat-transfer coefficient (based on bare-tube area), kcal/h•m²•°C	830.6	961.0
Air flow rate, kg/h	2,200,000	2,200,000
Air pressure drop, mm H₂O	13	12.219
Power consumed, kW	34.83	34.93
Tubeside pressure drop, kg/cm²	0.41	0.5
Degree of overdesign, %	4.15	4.16
Bare-tube area, m²	739	654
Fin area, m²	12,355	14,978
Total area (bare-tube + fin), m²	13,094	15,632

tubeside heat-transfer coefficient is controlling.

Example 5 (airside heat-transfer coefficient controlling). A closed-circuit tempered-water cooler was to be designed for a refinery application with the following parameters: flow rate = 220,000 kg/h (base flow rate of 200,000 kg/h multiplied by a safety factor of 1.1 to account for uncertainties in simulation, variations in feedstock quality, or capacity increases), inlet temperature = 120.8°C, outlet temperature = 60°C, allowable pressure drop = 0.5 kg/cm², fouling resistance = 0.0002 h•m²• C/kcal, heat duty = 13.376 MM kcal/h (12.16 MM kcal/h × 1.1), and design ambient temperature = 42°C. The tubes were made of carbon steel, had dimensions of 25 mm O.D. × 2.5 mm thick × 12,500 mm long, and had embedded (G-type) fins.

Two designs were prepared, one using ½-in.-high fins and the other using ⅝-in.-high fins; these are detailed in Table 4. The overdesign and power consumption were the same to allow a meaningful comparison to be made.

As expected, since the airside heat transfer controls, the design using ⅝-in.-high fins proved to be more economical. For the same overdesign and power consumption, the ½-in.-fin-height design had a significantly lower bare-tube area (11.5%). While it is true that the finned area was greater (21.2%), this represents a smaller increase in cost than the savings due to the lower tube cost. (The fabrication cost, which represents a major component of the cost of a finned tube, depends upon the length of tubing to be finned and does not vary to any significant extent with fin height.) Since the ⅝-in.-fin-height design requires the finning of 11.5% fewer tubes, the overall cost of this design will be less, as both the tube cost and the finning cost will be less.

An unusual situation arises sometimes where, although the tubeside heat-transfer coefficient is controlling, a design with higher fins proves to be more economical. This happens when the tubeside pressure drop is not fully utilized, and therefore the tubeside heat-transfer coefficient is not maxi-

mized. Any increase in the number of tube passes results in the tubeside pressure drop exceeding the allowable limit. For example, by switching from ½-in.-high to ⅝-in.-high fins, the number of tubes can be reduced by about 10%, thereby leading to roughly an 8% increase in the tubeside heat-transfer coefficient.

Example 6 (tubeside heat-transfer coefficient controlling). A vacuum-column bottoms cooler had to be designed for a refinery hydrocracker unit. The principal process parameters were: flow rate = 93,278 kg/h (84,798 kg/h × 1.1), inlet temperature = 180°C, outlet temperature = 70°C, inlet viscosity = 4.2 cP, outlet viscosity = 15 cP, heat duty = 5,764,000 kcal/h (5,240,000 × 1.1), allowable pressure drop = 1.6 kg/cm², fouling resistance = 0.0006 h•m²•°C/kcal, and design ambient temperature = 42°C. The specified tube size was 25 mm O.D., 2.5 mm thick, and 12,500 mm long: the tube material was carbon steel.

Because the tubeside viscosity was rather high, the tubeside heat-transfer coefficient was controlling. Therefore, a design was made with tubes having ½-in.-high fins and 276 fins/m (the following section on fin spacing explains why the maximum fin density of 433 fins/m was not used), as detailed in Table 5.

The airside heat-transfer resistance was only 12.2% of the total resistance, whereas the tubeside resistance was 82.48%.

Next an alternative design was prepared with tubes having ⅝-in.-high fins. The number of tubes per row was reduced from 50 to 45 so that the bundle width would be the same as in the original (the tube pitch in the second design was 67 mm vs. 60 mm for the design with ½-in.-high fins). The airside flow rate was adjusted to 1,660,000 kg/h to maintain the same power consumption. The tubeside pressure drop, 1.85 kg/cm², was excessive.

Reducing the number of tube passes to nine resulted in the design becoming undersurfaced by 3.84%. Unfortunately, the number of tubes per row could

not be increased, as the bundle width was already at the permitted maximum of 3.2 m. As expected (since the airside was not controlling), increasing the air flow rate made no significant improvement in performance.

Therefore, the only alternative was to increase the number of tube rows from eight to nine. The number of tube passes was kept at ten (two in the first row and one in each of the subsequent seven rows) and the air flow rate was reduced from 1,660,000 to 1,600,000 kg/h to have the same airside power consumption.

This design was acceptable, as it was 5.7% oversurfaced and the tubeside pressure drop was 1.39 kg/cm². However, while the bare-tube area was only marginally higher than that of the original design (1,554 m² vs. 1,532 m²), the finned area was considerably higher (24,037 m² vs. 17,504 m²). Consequently, the original design was superior.

Fin spacing

The logic described above for fin height is applicable here, too — a higher fin density is economically fa-

vorable for services where the airside heat-transfer coefficient is controlling. Fin spacing can be optimized while fine-tuning a design. It usually varies between 7 and 11 fins/in (276 to 433 fins/m). A higher fin density will result in a higher airside heat-transfer coefficient, as well as a higher airside pressure drop.

Example 7. For the tempered-water cooler discussed in Example 5, the variations in airside heat-transfer coefficient and pressure drop with fin density are shown in Table 6.

For a typical air mass velocity, the reduction in the airside heat-transfer coefficient is significantly more than that in the air pressure drop and power consumption. In other words, a higher fin density is more efficient at converting pressure drop to heat transfer. Therefore, whenever the airside heat-transfer coefficient is controlling, a higher fin density should be employed.

However, where the airside heat-transfer coefficient is not controlling, it is sensible to use a lower fin density to reduce power consumption. This is because a higher airside heat-transfer coefficient will not result in an apprecia-

Table 5. Original design for evaluating fin height in a tubeside-controlling application (Example 6).

	Fins: ½-in. high
Number of sections	2
Number of bundles per section	2
Number of tube rows	8
Number of tubes per row	50
Number of tube passes	10 (2 passes in each of the first 2 rows, 1 pass in each of the last 6 rows)
Air flow rate, kg/h	1,600,000
Air inlet temperature, °C	42
Air outlet temperature, °C	56.9
Bare-tube area, m²	1,536
Total area (bare-tube + fin), m²	17,942 m2
Fan diameter, m	4,572
Number of fans per section	2
Total power consumption, kW	19.3 kW/fan × 4 fans = 77.2
Tubeside pressure drop, kg/cm²	1.54

	11 fins/in. 433 fins/m	10 fins/in. 394 fins/m	9 fins/in. 354 fins/m	8 fins/in. 315 fins/m	7 fins/in. 276 fins/m
Table 6. Variation in airside heat-transfer coefficient and pressure drop with fin density (Example 7).					
Airside pressure drop, mm H$_2$O	12.219	11.63	11.039	10.472	9.911
Step-wise reduction in air pressure drop, %	—	4.8	5.1	5.1	5.4
Airside heat-transfer coefficient (based on bare-tube area), kcal/h•m²•°C	961.0	883.9	804.8	727.6	650.2
Step-wise reduction in airside heat-transfer coefficient, %	—	8.0	9.1	9.6	10.6
Power consumption, kW	34.93	33.68	32.43	31.22	30.03
Step-wise reduction in power consumption, %	—	3.58	3.71	3.73	3.81

ble increase in the overall heat-transfer coefficient, so the increased power consumption will not be justified.

One might argue that instead of using a lower fin density in such cases, a lower air mass-flow rate may be used along with a higher fin density so that the power consumption is the same. However, the lower air flow rate will result in an increase in the outlet air temperature and therefore a reduction in the MTD. This will result in an inferior design.

Number of tube rows

The minimum number of tube rows recommended to establish a proper air flow pattern is four, although three tube rows can also be used in exceptional circumstances while maintaining a higher overdesign. Most ACHEs have four to six tube rows, although eight or even ten rows may be used occasionally.

The advantage of using more tube rows is that more heat-transfer area can be accommodated in the same bundle width. This leads to a reduction in the number of tube bundles and sections, thereby reducing the plot area required for the ACHE and thus the cost.

The disadvantage is that in addition to increasing the fan horsepower (for the same air velocity), it also reduces the airside flow area for a given heat-transfer area and thereby the air flow rate itself, thus lowering the MTD. In

fact, this is one of the fascinating aspects of ACHE design — even the coolant flow rate is variable and has to be optimized. For water-cooled heat exchangers, the cooling water flow rate is virtually fixed, depending upon its inlet temperature and maximum outlet temperature.

The airside pressure drop increases directly with the number of tube rows and so does the heat-transfer area. However, as the number of tube rows increases, the airside velocity will have to be reduced for the same fan horsepower, and this will translate into a lower airside heat-transfer coefficient. This is why the number of tube rows has to be optimized carefully. For simplicity, this may be summarized by stating that the larger the number of tube rows, the lower will be the airside heat-transfer coefficient.

It follows, then, that when the tube-side heat-transfer coefficient is controlling, a larger number of tube rows may be used because the lower airside heat-transfer coefficient will not adversely affect the design (since it is not controlling). However, when the airside heat-transfer coefficient is controlling, the number of tube rows will have to be limited to a lower value for an optimum design.

Thus, for services such as condensers, water coolers, and light hydrocarbon liquid coolers having a high tubeside heat-transfer coefficient, the

optimum number of tube passes is likely to be four or five. However, for gas coolers and viscous liquid coolers, the optimum number of tube rows is likely to be six, seven, or even higher, with the maximum generally being eight to ten. This is because with a larger number of tube rows, the airside heat-transfer coefficient will be inordinately low (due to the low velocity sustainable for a normal power consumption) or the airside pressure drop will be inordinately high (if a normal airside velocity is used).

Fan power consumption

Fan power varies directly as the volumetric air flow rate and the pressure drop; the volumetric air flow rate varies directly as the air mass velocity, while the pressure drop varies to the 1.75 power of air mass velocity. Thus, air pressure drop varies to the 1.75 power and fan horsepower to the 2.75 power of the air mass velocity. In sharp contrast, the air heat-transfer coefficient varies to the 0.5 power of the air mass velocity. This is represented graphically in Figure 11.

Thus, when the air mass velocity is increased, the air pressure drop and the fan power increase rather sharply (especially the latter), whereas the airside heat-transfer coefficient increases at a much slower rate. It follows, therefore, that there will be an optimum air mass velocity beyond which any further increase will be wasteful. This optimum

air mass velocity will depend upon the extent to which the airside heat-transfer coefficient is controlling. In a situation where the airside heat-transfer coefficient is highly controlling, the optimum air mass velocity will be higher than in a situation where the airside heat-transfer coefficient is much less controlling.

Example 8. For convenience, the tube metal resistance (which is negligible) will be ignored, and the tubeside fouling resistance and tubeside film resistance will be combined into a single term called the tubeside resistance.

Let us first look at what happens when the airside is controlling. The airside heat-transfer coefficient (htc_a) = 800 kcal/h•m²•°C and the tubeside heat-transfer coefficient (htc_t) = 4,000 kcal/h•m²•°C; so, the overall heat-transfer coefficient (U) = 667 kcal/h•m²•°C. If the air flow rate is increased by 25%, htc_a will increase to 894.4 kcal/h•m2•°C, so that U will increase to 731 kcal/h•m2•°C, a difference of 9.6%. This has been achieved at the expense of a $(1.25)^{1.75}$, or 48%,

increase in air pressure drop and a $(1.25)^{2.75}$, or 85%, increase in fan power consumption.

Now let's consider the case where the tubeside is controlling, with htc_a = 800, htc_t = 250, and U = 190.5, all in units of kcal/h•m²•°C. The same 25% increase in air flow rate will again cause htc_a to increase to 894.4 kcal/h•m²•°C. Here, U will increase from 190.5 to 195.4 kcal/h•m²•°C, an increase of only 2.6%. As earlier, the airside pressure drop will increase by 48% and the fan power consumption by 85%.

Thus, for the same increase in pressure drop and fan power consumption, the increase in the overall heat-transfer coefficient is 9.6% when the airside is controlling but only 2.6% when the tubeside is controlling. Therefore, while this increase in air flow may be justifiable when the airside controls (depending upon the actual heat-transfer area, air pressure drop, and fan power consumption), it is certainly unlikely to be justifiable when the tubeside controls.

Tube pitch

Although tubes can be laid out in either a staggered or an in-line pattern, the former is almost invariably employed because it results in much better performance in terms of conversion of pressure drop to heat transfer. Tube pitch has a very profound effect on airside performance. The transverse pitch is more crucial and is what is implied by the term "tube pitch." The longitudinal pitch has much less influence and is usually 80–90% of the transverse pitch, in order to minimize the height of the tube bundles and thereby the cost.

Designers tend to use the following standard combinations of bare-tube O.D., finned-tube O.D., and tube pitch, as they tend to be optimum:

1-in./2-in./2.375-in.
(25-mm/50-mm/60-mm)

1-in./2.25-in./2.625-in.
(25-mm/57-mm/67-mm)

However, in many situations, these may not be the optimum. So, the tube pitch should be varied and the optimum established for each application.

The normal range of tube pitch for the 1-in./2-in. (bare-tube O.D./finned-tube O.D.) combination is 2.125 in. to 2.5 in. Similarly, for a 1-in./2.25-in. combination, the normal range of tube pitch is 2.375 in. to 2.75 in.

At a relatively lower tube pitch, the air pressure drop and therefore the power consumption tend to be high for the same airside heat-transfer coefficient. In other words, as the tube pitch is decreased, the airside pressure drop and power consumption increase more rapidly than does the airside heat-transfer coefficient.

This is illustrated by the results of a study summarized in Table 7. The tubes were 1-in.-O.D. tubes with ½-in.-high fins; four tube pitches — 2.125 in., 2.25 in., 2.375 in., and 2.5 in. — and four fin densities — 5 fins/in. (fpi), 7 fpi, 9 fpi, and 11 fpi — were evaluated. The combination of 2.625 in. tube pitch and 11 fpi was consid-
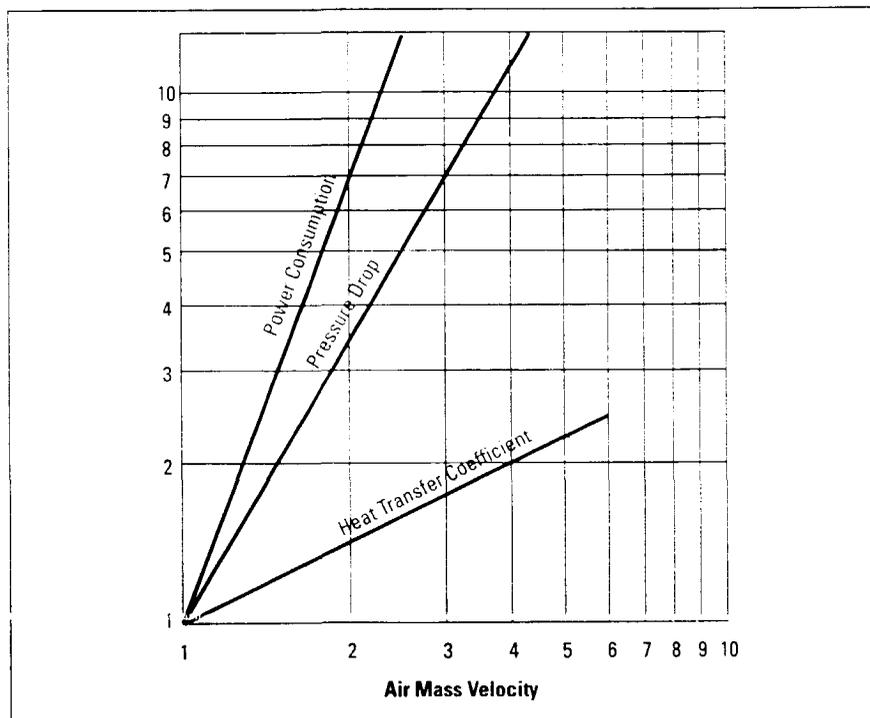
■ *Figure 11. Variation of airside heat-transfer coefficient, pressure drop, and power consumption with air mass velocity.*

Table 7. Effect of tube pitch and fin density on heat-transfer coefficient (HTC) and pressure drop (PD).

	5 fins/in. 197 fins/m		7 fins/in 276 fins/m		9 fins/in. 354 fins/m		11 fins/in. 433 fins/m	
	PD	HTC	PD	HTC	PD	HTC	PD	HTC
Tube Pitch = 2.125 in.	74.2	61.3	91.7	76.4	109.8	90.6	129.5	104.7
Tube Pitch = 2.25 in.	65.9	59.9	81.1	74.5	96.2	88.7	112.9	101.9
Tube Pitch = 2.375 in.	59.1	59.0	72.7	73.1	86.4	86.8	100.0	100.0
Tube Pitch = 2.5 in.	53.8	58.0	65.9	72.2	77.3	85.4	89.4	98.1

ered the base case (since this a very common configuration), and 100 units were assigned for both heat-transfer coefficient (HTC) and pressure drop (PD). The performances of the other combinations, in terms of heat-transfer coefficient and pressure drop, were determined relative to the base case.

As expected, the reduction in pressure drop was much sharper than the reduction in heat-transfer coefficient. Typically, for each step change in tube pitch, the pressure drop decreased by about 10%, whereas the heat-transfer coefficient decreased by only about 2%. This pattern was essentially the same for all the various fin densities.

The designer may, therefore, feel that the higher the tube pitch, the better the performance of an ACHE. While this is basically true, there is another extremely important factor that should not be overlooked: As the tube pitch is increased for a given number of tubes, the tube bundle width will increase (even the fan diameter will increase at a certain point), thus pushing up the cost of the ACHE. Consequently, the tube pitch should be optimized in the *overall* context.

While the optimum tube pitch may vary from situation to situation, designers typically prefer to use a 2.375-in. tube pitch for a 1-in./2-in. combination, as this generally represents the best compromise between the two opposing tendencies of performance efficiency and cost. For a 1-in./2.25-in. combination, the following tube pitches may be used — 2.375 in., 2.5 in., 2.625 in.,

and 2.75 in, with 2.625 in. being the most common.

It is important to realize that tube pitch is a powerful variable in ACHE design. It can be fine-tuned in very small steps — as little as 1 mm (and not necessarily ⅛-in. increments), depending upon the situation at hand. For example, tube pitch can be decreased to accommodate the required number of tubes per row within the maximum permitted tube bundle width (which is determined by transportation constraints) at the expense of a somewhat higher power consumption. Similarly, it can be decreased or increased to allow the use of a standard fan diame-

ter in order to meet the API 661 stipulation of a minimum fan coverage of 40%.

Number of tube passes

The distribution of tubes in the various passes need not be uniform. For instance, an ACHE with six tube rows can have four tube passes with two rows in each of the upper two passes and one row in each of the lower two tube passes.

This feature is especially useful in condensers — the flow area in each pass can be gradually reduced as the liquid fraction (and therefore the mixture density) increases progressively, thereby obtaining a more uniform pres-

Table 8. Variation of performance with tube-pass distribution in a two-pass condenser (Example 9).

	1st pass, 4 rows; 2nd pass, 1 row	1st pass, 3 rows; 2nd pass, 2 rows	1st pass, 2.5 rows; 2nd pass, 2.5 rows	1st pass, 2 rows; 2nd pass, 3 rows
Tubeside heat-transfer coefficient, kcal/h•m²•°C	2,309.5	2,576	2,673	2,732
Overall heat-transfer coefficient, kcal/h•m²•°C	480.8	493.4	497.6	499.9
Mean temperature difference, °C	38.63	39.9	39.92	39.59
Degree of overdesign, %	0.82	6.88	7.84	7.45
Tubeside pressure drop, kg/cm²	0.483	0.496	0.634	0.919
Ratio of tubeside heat-transfer coefficient to pressure drop	4,782	5,194	4,216	2,973

Note: An additional design, with one row in the first pass and four rows in the second pass, was not feasible because the tubeside pressure drop was greater than the operating pressure.

Table 9. Process parameters, and construction and performance details, for Example 10, a combined service.

Service	ACHE #1 Stripped Light Cycle Oil Cooler	ACHE #2 Stripped Heavy Naphtha Cooler	ACHE #3 Circulating Heavy Naphtha Cooler
Flow rate, kg/h	48,949	19,216	74,114
Heat duty, MM kcal/h	2.48	1.02	1.64
Inlet temperature, °C	210	165	177
Outlet temperature, °C	66	66	138
Operating pressure, kg/cm² g abs.	9.8	9.4	8.8
Allowable pressure drop, kg/cm²	0.7	0.5	0.7
Fouling resistance, h•m²•°C/kcal	0.0004	0.0003	0.0003
Inlet viscosity, cP	0.4	0.3	0.28
Outlet viscosity, cP	2.66	0.7	0.36
Number of tubes per row	48	15	12
Number of rows	6	6	6
Number of tube passes	6	4	2
Approximate bundle width, m	3.0	1.0	0.8
Air flow rate, kg/h	370,700	115,800	88,700
Air inlet temperature, °C	42	42	42
Air outlet temperature, °C	69.9	72.6	119.0
Static pressure, mm w.g.	13.0	13.0	13.0
Tubeside pressure drop, kg/cm²	0.7	0.2	0.6
Mean temperature difference, °C	52.8	50.9	73.9
Heat-transfer area (bare), m²	235	73.5	58.8

sure drop in the various passes than would be possible in an even distribution of 1.5 tube rows in all passes. This results in a higher overall pressure-drop-to-heat-transfer conversion, which in turn results in a lower heat-transfer area.

This can be restated as follows: If the number of tubes per pass were to be identical in all the passes, the initial passes (where there is more vapor and consequently a lower density) will have a higher pressure drop without a correspondingly higher heat-transfer coefficient. The subsequent tube passes will have much lower pressure drops. The design will, therefore, not be optimum with respect to the best utilization of available pressure drop.

Example 9. An atmospheric-column overhead condenser using air as the cooling medium was to be de-

signed for the crude distillation unit of a refinery based on the following parameters: heat duty = 67.98 MM kcal/h (56.65 × a safety factor of 1.2), overhead vapor flow rate = 501,744 kg/h (418,120 kg/h × 1.2) (hydrocarbon = 95.8 %wt. and steam = 4.2 %wt.), inlet temperature = 144.6°C, outlet temperature = 65°C, weight fraction condensed (at outlet) = 0.995, operating pressure = 3.7 kg/cm² abs., allowable pressure drop = 0.5 kg/cm², fouling resistance = 0.0002 h•m²• C/kcal, and design ambient temperature = 42°C.

An exchanger was designed with eight sections and two tube bundles per section. Each tube bundle had 48 tubes per row and five tube rows. Tubes were made of carbon steel, 25 mm O.D., 2.5 mm thick, and 12,500 mm long, with 12.5-mm-high aluminum fins spaced at 433 fins/m. Several possible tube pass

arrangements were examined, as summarized in Table 8.

The design having three tube rows in the first (upper) pass and two rows in the second (lower) pass had the highest efficiency for converting pressure drop to heat transfer. Thus, this is the optimum design.

It should be noted that the variation in tubeside heat-transfer coefficient among the different designs is nominal. On the other hand, the variation in tubeside pressure drop is considerable.

It should also be noted that having a full row of tubes in one pass results in a better tubeside distribution than having fractional rows per pass. Thus, for a 6-row/4-pass design, it is better to have two passes of two rows each and two passes of one row each than to have four passes of 1.5 rows each.

Special applications

We will conclude by discussing several special applications — combined services, recirculation ACHEs, humidified ACHEs, tube inserts, variable finning density, natural convection, and air-cooled vacuum steam condensers.

Combined services

A difficulty arises when several small coolers need to be located on a broad piperack, say, 8 m wide. Because the heat-transfer areas are rather small, single tube bundles of narrow width (1 m to 3 m) may suffice. In these cases, it is a common practice to combine several such coolers in a single section. For example, three tube bundles of widths 1 m, 2 m, and 3 m may be combined to form a section a little over 6 m wide: two tube bundles for one service may be combined with one tube bundle for another service to form a section; and so on. This is illustrated by the following example.

Example 10. Three ACHEs were to be designed for a fluid catalytic cracking (FCC) unit of an oil refinery using a design ambient temperature of 42°C and the process parameters detailed in Table 9.

Carbon steel tubes, 25 mm O.D. × 2.5 mm thick × 10,500 mm long, were

to be used, as the ACHEs were to be mounted on a 10-m-wide piperack. Since the heat duties were rather small, a combined design was prepared wherein three tube bundles (one for each service) were combined in a single section, using G-finned tubes with 12.5-mm-high aluminum fins, spaced at 43 fins/m, at a transverse pitch of 60 mm, as elaborated in Table 9. Two fans, 3.657 m in diameter, were selected to force the total air flow rate of 575,200 kg/h against a static pressure of 13 mm w.g., consuming 16.4 kW each; two motors of 18.75 kW each were selected to drive the fans. Split headers were used to handle all three bundles, since all of them had design temperatures in excess of 200°C and the first two had a high temperature difference (between the inlet and the outlet temperatures of the process fluid) as well.

It may be noted that the air outlet temperature from ACHE #3 was rather high, at 119°C. This was a direct consequence of the high MTD, which required a relatively small heat-transfer area and therefore a rather small face area, through which only a relatively low flow rate of air could be passed.

The disadvantage in combining different services in one section is the loss of individual control. Consequently, this practice is not recommended for condensers, but is limited to product coolers.

Recirculation ACHEs

There are situations where a minimum tubewall temperature must be maintained. An example is the overhead condenser for a sour water stripper, where if the tubewall temperature falls below 70°C, solidification fouling takes place due to the deposition of ammonium salts. Since the ambient temperature varies from day to night and from season to season, a special arrangement is needed to maintain a constant air temperature (the design ambient temperature) at all times to the fans delivering air to the condenser tube bundles (forced draft).
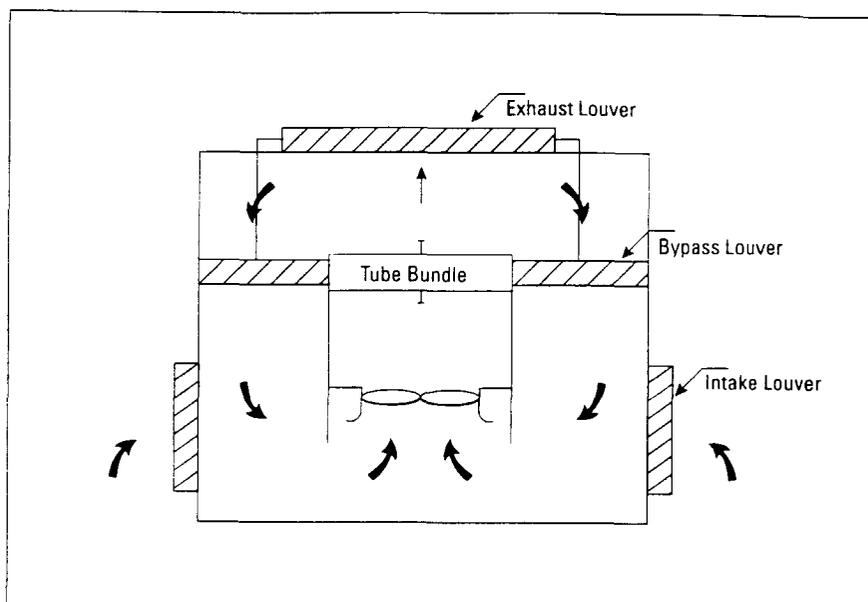
■ Figure 12. Automatic louvres on recirculation ACHEs maintain a constant air temperature to the fans.

Such a situation can employ a recirculation ACHE, wherein automatic louvers at the top and side(s) of the housing control the extent of recirculation. As shown in Figure 12, a part of the air leaving from the top of the tube bundles is recirculated and mixes with fresh air entering from the side(s), so that the combined temperature is precisely the design ambient temperature. Obviously, the lower the ambient temperature, the greater will be the extent of recirculation. Automatically controlled louvers (sensing the air temperature just below the fans) at both the top and the side(s) ensure the desired air temperature at all times. This arrangement can also be used for cooling heavy crude stocks with high pour points.

One difficulty with recirculation ACHEs will be startup during winter, or even summer evenings if the ambient temperature falls below the design ambient temperature. This problem can be overcome by placing steam coils below the tube bundles — the cold ambient air is warmed up to the design ambient temperature by passing low-pressure steam through the coils and having total air recirculation until the air temperature reaches the design ambient temperature. The steam supply

can then be stopped, as the ACHE will be able to take care of itself.

Humidified ACHEs

Normal air-cooled heat exchangers can cool a process fluid only to a temperature that is higher than the dry-bulb temperature of the ambient air. Therefore, in hot arid areas, the utility of air cooling is severely limited.

However, by humidifying the hot dry air, the temperature can be brought down considerably, thereby extending the capability of air cooling significantly.

Humidified ACHEs are normally of the induced-draft type. Water is sprayed into the air stream before it comes in contact with the heat exchanger surface, thereby lowering the air temperature due to evaporation. Mist eliminators are needed to prevent droplets of water from entering the tube bundle. The water that does not evaporate is collected in a basin beneath the cooler and recirculated.

The installed cost of humidified ACHEs is rather high. And, the cost of makeup water represents an additional expense. Another limitation of humidified air cooling is that soft water must be used, otherwise there will be scaling

of the finned tubes with consequent deterioration in cooler performance.

However, since humidification enhances the cooling capability considerably, a proper economic study will have to be carried out to establish the viability of this type of exchanger. It is important to remember that in arid areas, the scarcity of water may necessitate this mode of cooling for services with streams that must be cooled to temperatures below summer day temperatures.

An important factor to consider is that humidification of the coolant air will be required only during the daytime in the summer, which may represent only 4–5% of the total operation of the equipment. Certainly, the selection of the design ambient temperature will have to be done very carefully.

Tube inserts

Tube inserts considerably enhance the tubeside heat-transfer coefficient under laminar flow conditions. The first-generation tube inserts were twisted tapes that imparted a swirling motion to the tubeside fluid, thereby augmenting heat transfer. The second-generation tube inserts are wire-fin tube inserts (Figure 13) that increase turbulence dramatically by promoting radial mixing from the tubewall to the center of the tubes, thereby eliminating the boundary-wall problem. By virtue of the increased tubeside (and thereby overall) heat-transfer coefficient, wire-fin tube inserts are very useful in ACHEs cooling viscous liquids, especially in an offshore platform where floor space is at a premium.

Example 11. Three coolers were to be designed for an offshore platform and incorporated in a plot area of 2.2 m × 7.0 m. The process parameters are detailed in Table 10. The design ambient temperature was 40°C.

The original design used carbon steel tubes that measured 1 in. O.D. × 12 BWG (minimum, under groove). Due to the rather high viscosity, laminar flow in all three coolers resulted in extremely low tubeside heat-transfer coefficients. Consequently, the required plot area was

Table 10. Process parameters and thermal design for Example 11, designing three coolers for an offshore platform			
Fluid handled	**ACHE #1** **Synthetic oil**	**ACHE #2** **Turbine oil**	**ACHE #3** **Turbine oil**
Flow rate, kg/h	2,074	7,438	35,449
Inlet temperature, °C	74	66	75
Outlet temperature, °C	54	54	54
Operating pressure, kg/cm² abs.	6.2	46.0	7.0
Allowable pressure drop, kg/cm²	0.7	1.0	1.0
Fouling resistance, h•m²•°C/kcal	0.0004	0.0004	0.0004
Heat duty, kcal/h	19,030	44,440	373,670
Inlet viscosity, cP	8.3	10	9.2
Outlet viscosity, cP	14	21	21
Number of tube bundles	1	1	1
Number of tubes per row	2	4	23
Number of tube rows	7	7	7
Bundle width, m	0.2	0.34	1.47
Number of tube passes	2	1	1
Tubeside pressure drop, kg/cm²	0.7	0.91	1.0
Heat-transfer area (bare), m²	7.2	14.3	82.3
Air flow rate, kg/h	13,000	26,000	149,530
Inlet air temperature, °C	40	40	40
Outlet air temperature, °C	46.1	47.1	50.4
Mean temperature difference, °C	19.9	15.0	17.4
Overall heat-transfer coefficient, kcal/h•m²•°C	133	207	261

considerably higher than the allowable value, and the tubeside pressure drops exceeded the allowable limits.

Therefore, revised designs were attempted incorporating wire-fin inserts in tubes that were 6.5 m long and had ½-in.-high aluminum fins spaced at 433 fin√m. A combined section using one tube bundle for each service could be easily accommodated within the specified plot area (the actual plot area was 2.01 m × 6.5 m). The features of the thermal design are detailed in Table 10. Three fans, 1.524 m in diameter, were employed to drive the total air flow of 188,530 kg/h across the three tube bundles.

It should be highlighted here that the overall heat-transfer coefficients of 133, 207, and 261 kcal/h•m²•°C obtained with wire-fin tube inserts are

order-of-magnitude times higher than what can be achieved with bare tubes.

Variable finning density

Variable finning density can be useful for the cooling of liquids with viscosity that changes considerably from inlet to outlet. With the increase in viscosity, the tubeside heat-transfer coefficient decreases as the fluid flows from top to bottom. As mentioned previously, a higher finning density is favorable when the airside heat-transfer coefficient is controlling and a lower finning density is favorable when the tubeside heat-transfer coefficient is controlling. Thus, as the tubeside heat-transfer coefficient decreases, the tubeside resistance becomes more and more controlling and a lower finning density becomes more favorable.

Consequently, when cooling a viscous liquid, it is often advantageous to have a higher finning density in the upper rows and a lower finning density in the lower rows. The degree of overdesign may be slightly less, but there will be an appreciable savings in power consumption. Furthermore, the greater the variation in viscosity of the tubeside liquid, the greater is the potential for this type of variable finning.

The vacuum-column bottoms cooler discussed in Example 6 earlier had two sections, each having two tube bundles, and each bundle having 50 tubes per row, eight rows, and ten tube passes. The fin density was 276 fins/m. The effect of alternative finning can be evaluated by keeping the fin density at 276 fins/m in the top six tube rows and changing it to 197 fins/m in the lower two tube rows. The fan power requirement drops from 18.92 kW to 18.4 kW,

whereas the degree of overdesign falls from 8.85% to 8.25%. Switching to 276 fins/m in the upper four rows and 197 fins/m in the lower four rows reduces the fan power requirement further to 17.88 kW and reduces the overdesign to 7.56%. Thus, for a 1.2% reduction in overdesign, the power consumption is cut by 5.8%.

Natural convection

In cases where the heat duty is very small and the tubeside heat-transfer coefficient is rather low, no fans need be used, so that power consumption is saved (15). This is illustrated by the following example.

Example 12. A vacuum residue cooler for a refinery was to be designed based on the following data: heat duty = 476,400 kcal/h, flow rate = 17,604 kg/h, inlet temperature = 267°C, outlet temperature = 220°C, inlet viscosity = 18 cP, outlet viscosity = 25.3 cP, fouling resistance = 0.002 $h \cdot m^2 \cdot °C/kcal$, and allowable pressure drop = 1.0 kg/cm^2. Carbon steel tubes, 25 mm O.D. × 2.5 mm thick × 6.0 m long, were to be used.

A natural-draft ACHE was designed as follows. It had one section having one tube bundle, 40 tubes/row, four tube rows, and eight tube passes. Fin height was 12.5 mm, transverse tube pitch was 60 mm, and fin density was 433 fins/m. A 3-m-high stack was incorporated.

The tubeside and airside heat-transfer coefficients were 61.53 kcal/h·m²· C and 239.8 kcal/h·m²· C, representing 70.4% and 18.1% of the overall resistance, respectively. The airside pressure drop was only 0.576 mm w.g., whereas the tubeside pressure drop was 0.74 kg/cm^2. The total bare-tube area was only 66.0 m^2. Since the air flow rate was only 24,267 kg/h, the air outlet temperature was rather high, at 118°C. Louvers were provided to cut off the air flow (even though it was small) when the unit was not in use, so as to prevent congealing of the vacuum residue.

It should be noted that since the tubeside heat-transfer coefficient was extremely low, the use of fans would

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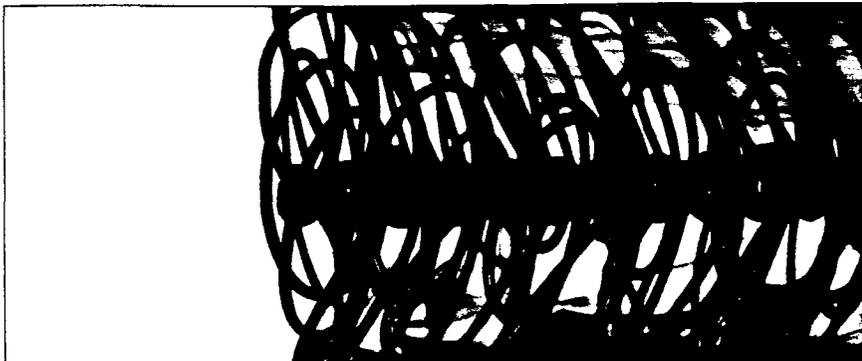
■ *Figure 13. Wire-fin tube inserts improve heat transfer by increasing turbulence. (Photo courtesy of Cal Gavin, Ltd.)*

have resulted in a negligible increase in the overall heat-transfer coefficient. However, the cost would have been significantly higher because of the costs for the fans, drives, plenums, and so on.

Air-cooled vacuum steam condensers

Steam turbines are very widely used in the chemical process industries for driving not only electric generators, but also various types of pumps, fans, compressors, and other equipment. Steam condensers are required to condense the exhaust of these turbines and return the condensate to the boiler. This condenser can either be water-cooled or air-cooled. (The advantages and disadvantages of each are the same as elaborated at the beginning of the article.) A typical A-frame air-cooled vacuum steam condenser (ACVSC) is shown in Figure 9.

The main problem with the ACVSC is not the condensation of the steam, but the evacuation of the noncondensables. (The noncondensables are the gases that enter the vacuum section of the power cycle from the atmosphere as well as from the chemicals used for the treatment of boiler feedwater.) Fail-

ure to eliminate the noncondensables can cause freezing of condensate in winter, loss of performance due to blanketing of the heat-transfer surface, and absorption of noncondensables by condensate and subsequent corrosion of tube metal. Thus, a successful ACVSC must continuously and totally collect and eliminate all noncondensables from the system.

The trapping of noncondensables inside the condenser tubes is a direct consequence of the variation of coolant air temperature across the tube bundle. Consider a single-pass condenser having four tube rows. The tubes of the lowest row are exposed to the coldest air while the tubes of the upper rows are exposed to progressively hotter air. Therefore, the tubes in the lowest row condense more steam (due to the higher MTD), while those in the upper rows condense less and less steam. Consequently, the pressure drop will be the highest in the tubes in the lowest row and progressively lower in the tubes of the upper rows. This will cause a backflow of noncondensables from the tubes of the upper rows to the tubes of the lowest row, which can eventually lead to gas blanketing of a substantial fraction of the heat-transfer surface. This problem will be less acute in, but not absent from, a 2-pass/4-row construction.

In order to address this situation, a correction factor has been proposed by Rozenman *et al. (16)*, wherein extra heat-transfer area is incorporated as per the penalty factor evaluated.

Further Reading

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Special, patented designs have been developed by some ACHE vendors. For a detailed presentation of the problem caused by incomplete evacuation of noncondensables, the reader should consult *(16, 17)*. **CEP**

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