

SECTION 10

Air-Cooled Exchangers

An air-cooled exchanger (ACHE) is used to cool fluids with ambient air. Several articles have been published describing in detail their application and economic analysis. (See Bibliography at the end of this section.) This section describes the general design of air-cooled exchangers and presents a method of approximate sizing.

ARRANGEMENT AND MECHANICAL DESIGN

Figs. 10-2 and 10-3 show typical elevation and plan views of horizontal air-cooled exchangers as commonly used. The basic components are one or more tube sections served by one or more axial flow fans, fan drivers, speed reducers, and an enclosing and supporting structure.

Air-cooled exchangers are classed as forced draft when the tube section is located on the discharge side of the fan, and as induced draft when the tube section is located on the suction side of the fan.

Advantages of induced draft are:

- Better distribution of air across the section.
- Less possibility of the hot effluent air recirculating around to the intake of the sections. The hot air is discharged upward at approximately $2^{1/2}$ times the velocity of intake, or about 1500 ft/min.
- Less effect of sun, rain, and hail, since 60% of the face area of the section is covered.

FIG. 10-1
Nomenclature

A = area of heat transfer surface, sq ft	h_s = shell side film coefficient based on outside tube area, $\text{Btu}/(\text{h} \cdot \text{sq ft} \cdot ^\circ\text{F})$
A_i = inside surface of tube, sq ft	h_t = tube side film coefficient based on inside tube area, $\text{Btu}/(\text{h} \cdot \text{sq ft} \cdot ^\circ\text{F})$
A_b = outside bare tube surface, sq ft	HP = fan horsepower
A_x = outside extended surface of tube, sq ft	J = J factor (see Fig. 10-13)
A_t = tube inside cross-sectional area, sq in. (see Fig. 9-25)	k = thermal conductivity, $\text{Btu}/(\text{hr} \cdot \text{sq ft} \cdot ^\circ\text{F}/\text{ft})$
ACFM = actual cubic feet per minute	L = length of tube, ft
APF = total external area/ft of fintube, sq ft/ft	LMTD = log mean temperature difference, $^\circ\text{F}$ (see Fig. 9-3)
APSF = external area of fintube, sq ft/sq ft of bundle face area	N = number of rows of tubes in direction of flow
AR = area ratio of fintube compared to the exterior area of 1 in. OD bare tube	N_f = number of fans
B = correction factor, psi (see Fig. 10-14)	N_P = number of tube passes
C_p = specific heat at average temperature, $\text{Btu}/(\text{lb} \cdot ^\circ\text{F})$	N_R = modified Reynolds number, $(\text{in} \cdot \text{lb}/(\text{sq ft} \cdot \text{s} \cdot \text{cp}))$
CMTD = corrected mean temperature difference, $^\circ\text{F}$	N_t = number of tubes
dB(A) = overall weighted level of sound at a point distant from noise source based on "A" weighting system	ΔP = pressure drop, psi
D = fan diameter, ft	PF = fan total pressure, inches of water
D_i = inside tube diameter, in.	ρ_a = density of air, lb/cu ft
D_o = outside tube diameter, in.	ρ_w = density of water, lb/cu ft
D_R = density ratio, the ratio of actual air density to the density of dry air at 70°F and 14.7 psia, 0.0749 lb/cu ft (see Fig. 10-16)	P = temperature ratio (see Fig. 10-8)
f = friction factor (see Fig. 10-15)	PWL = sound pressure level
F = correction factor (see Fig. 10-8)	PWLN = PWL for N_f fans
F_a = total face area of bundles, sq ft	Q = heat transferred, Btu/h
F_p = air pressure drop factor, in. of water per row of tubes	r_d = fouling resistance (fouling factor), $(\text{hr} \cdot \text{ft}^2 \cdot ^\circ\text{F}/\text{Btu})$
FAPF = fan area per fan, ft^2/fan	r_f = fluid film resistance (reciprocal of film coefficient)
FPM = fan tip speed, feet per minute	r_{mb} = metal resistance referred to outside bare surface
g = local acceleration due to gravity, ft/s^2	r_{mx} = metal resistance referred to outside extended surface
G = mass velocity, $\text{lb}/(\text{sq ft} \cdot \text{s})$	R = distance in feet (see Equation 10-6)
G_a = air face mass velocity, $\text{lb}/(\text{hr} \cdot \text{sq ft})$ of face area	R = temperature ratio (see Fig. 10-8)
G_t = tubeside mass velocity, $\text{lb}/(\text{sq ft} \cdot \text{s})$	RPM = fan speed, rotations per minute
h_a = air side film coefficient $\text{Btu}/(\text{h} \cdot \text{sq ft} \cdot ^\circ\text{F})$	S = specific gravity (water = 1.0)
	SPL = sound pressure level
	t = temperature air side, $^\circ\text{F}$

FIG. 10-1 (Cont'd)
Nomenclature

T = temperature tube side, °F
 U = overall heat transfer coefficient, Btu/(h · ft² · °F)
 W = mass flow, lb/hr
 Y = correction factor, psi/ft (see Fig. 10-14)
 Δt = temperature change, °F
 μ = viscosity, cp
 μ_w = viscosity at average tube wall temperature, cp
 ϕ = viscosity gradient correction

Subscripts:

a = air side
 b = bare tube surface basis
 s = shell side
 t = tube side
 x = extended tube surface basis
 1 = inlet
 2 = outlet

- Increased capacity in the event of fan failure, since the natural draft stack effect is much greater with induced draft.

Disadvantages of induced draft are:

- Higher horsepower since the fan is located in the hot air.
- Effluent air temperature should be limited to 200°F, to prevent potential damage to fan blades, bearings, V-belts, or other mechanical components in the hot air stream.
- The fan drive components are less accessible for maintenance, which may have to be done in the hot air generated by natural convection.
- For inlet process fluids above 350°F, forced draft design should be used; otherwise, fan failure could subject the fan blades and bearings to excessive temperatures.

Advantages of forced draft are:

- Slightly lower horsepower since the fan is in cold air. (Horsepower varies directly as the absolute temperature.)
- Better accessibility of mechanical components for maintenance.
- Easily adaptable for warm air recirculation for cold climates.

The disadvantages of forced draft are:

- Poor distribution of air over the section.
- Greatly increased possibility of hot air recirculation, due to low discharge velocity from the sections and absence of stack.
- Low natural draft capability on fan failure due to small stack effect.
- Total exposure of tubes to sun, rain, and hail.

The horizontal section is the most commonly used air cooled section, and generally the most economical. For a fluid with freezing potential, the tubes should be sloped at least $\frac{1}{8}$ in. per

FIG. 10-2

Typical Side Elevations of Air Coolers

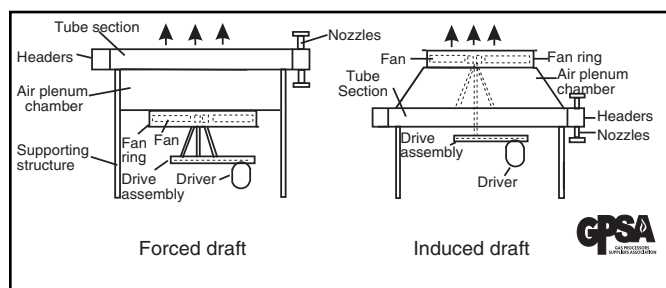
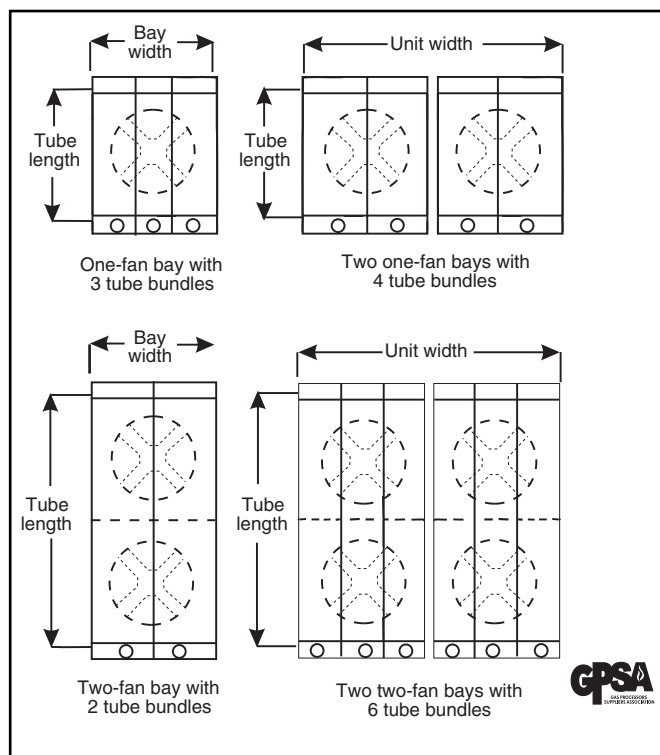


FIG. 10-3

Typical Plan Views of Air Coolers



foot to the outlet header. Since in most cases there will be no problem associated with freezing, and it is more costly to design a sloped unit, most coolers are designed with level sections.

Vertical sections are sometimes used when maximum drainage and head are required, such as for condensing services.

Angled sections, like vertical sections, are used for condensing services, allowing positive drainage. Frequently, angle sections are sloped thirty degrees (30°) from the horizontal. A-frames are usually sloped sixty degrees (60°) from the horizontal. See Fig. 10-4.

Fan sizes range from 3 ft to 28 ft diameter. However, 14 ft to 16 ft diameter is the largest diameter normally used. Fan drivers may be electric motors, steam turbines, hydraulic motors, or gas-gasoline engines. A speed reducer, such as a V-belt drive or reduction gear box, is necessary to match the driver output speed to the relatively slow speed of the axial flow fan. Fan tip speeds are normally 12,000 ft/min or less. General practice is to use V-belt drives up to about 50 bhp and gear drives at higher power. Individual driver size is usually limited to 50 hp.

Two fan bays are popular, since this provides a degree of safety against fan or driver failure and also a method of control by fan staging. Fan coverage is the ratio of the projected area of the fan to the face of the section served by the fan. Good practice is to keep this ratio above 0.40 whenever possible because higher ratios improve air distribution across the face of the tube section. Face area is the plan area of the heat transfer surface available to air flow at the face of the section.

The heat-transfer device is the tube section, which is an assembly of side frames, tube supports, headers, and fin tubes. Aluminum fins are normally applied to the tubes to provide an extended surface on the air side, in order to compensate for the relatively low heat transfer coefficient of the air to the tube. Fin construction types are tension-wrapped, embedded, extruded, and welded.

Tension-wrapped is probably the most common fin type used because of economics. Tension wrapped tubing is common for continuous service with temperatures below 400°F. Extruded fin is a mechanical bond between an inner tube exposed to the process and an outer tube or sleeve (usually aluminum) which is extruded into a high fin. Embedded fin is an aluminum or steel fin grooved into the base tube. Embedded fins are used in cyclic and high temperature services. Other types of finned tubes available are soldered, edge wrapped, and serrated tension wrapped. Coolers are regularly manufactured in tube lengths from 6 ft to 50 ft and in bay widths from 4 ft to 30 ft. Use of longer tubes usually results in a less costly design compared to using shorter tubes.

Base tube diameters are $\frac{5}{8}$ in. to $1\frac{1}{2}$ in. OD with fins from $\frac{1}{2}$ in. to 1 in. high, spaced from 7 to 11 per inch, providing an extended finned surface of 12 to 25 times the outside surface of the base tubing. Tubes are usually arranged on triangular pitch with the fin tips of adjacent tubes touching or separated by from $\frac{1}{16}$ in. to $\frac{1}{4}$ in. Matching of the tube section to the fan system and the heat transfer requirements usually results in the section having depth of 3 to 8 rows of fin tubes, with 6 rows the most typical.

A 1-in. OD tube is the most popular diameter, and the most common fins are $\frac{1}{2}$ in. or $\frac{5}{8}$ in. high. The data presented in Fig. 10-1 are for 1 in. OD tubes with $\frac{1}{2}$ in. high fins, 9 fins/in. (designated as $\frac{1}{2} \times 9$) and $\frac{5}{8}$ in. high fins, 10 fins/in. (designated as $\frac{5}{8} \times 10$).

Common materials of construction for headers are firebox quality carbon steel, ASTM SA-515-70, SA-516-70. Tubes are generally ASTM SA-214 (ERW), SA-179 (SMLS), carbon steel. Louvers are generally carbon steel, or aluminum with carbon steel construction being the most general and most economical. Fins are normally aluminum. Both stainless and brass alloys have their applications but are more expensive than carbon steel.

Inadvertent hot air recirculation by air cooled heat exchangers can reduce the performance by increasing the air inlet temperature to the bundle. Recirculation of hot air back to the cooler's inlet or to the inlet of another cooler can occur when coolers are located too close to each other or too close to large obstructions such as buildings. Arranging multiple coolers in a row, side by side, is often the best way to minimize inadvertent hot air recirculation. Forced draft coolers are more susceptible to recirculation than induced draft coolers. Refer to "Hot Air Recirculation by Air Coolers" by A. Y. Gunter and K. V. Shipes for more information.⁹ In some cases the layout of coolers and other items in a plant should be analyzed by Computational Fluid Dynamics methods to evaluate the degree of recirculation and the effect on plant performance.

HEADER DESIGN

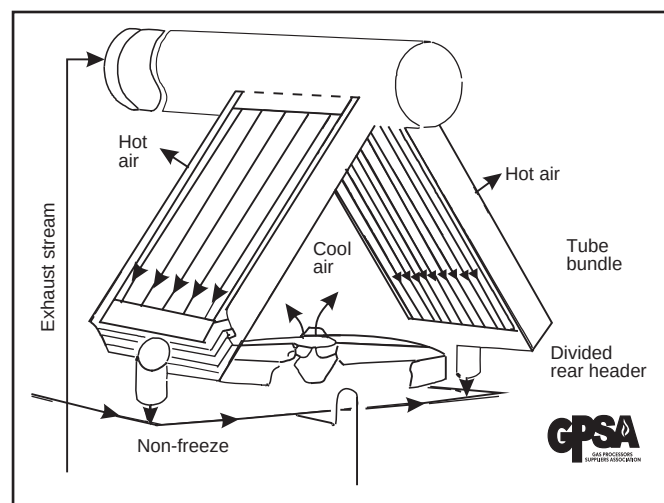
Plug header construction uses a welded box which allows partial access to tubes by means of shoulder plugs opposite the tubes. Plug headers are normally used as they are cheaper than the alternate cover plate design. Cover plate header construction allows total access to header, tube sheet, and tubes. This design is used in high fouling, low pressure service.

Fig. 10-5 shows typical designs for both plug header and cover plate header.

AIR-SIDE CONTROL

Air-cooled exchangers are sized to operate at warm (summer) air temperatures. Seasonal variation of the air temperature can result in over-cooling which may be undesirable. One way to control the amount of cooling is by varying the amount of air flowing through the tube section. This can be accomplished by using multiple motors, 2-speed drives, variable frequency drives (VFD), louvers on the face of the tube section, or variable pitch fans.

FIG. 10-4
Angled Section Layout



Staging of fans or fan speeds may be adequate for systems which do not require precise control of process temperature or pressure. Louvers will provide a full range of air quantity control. They may be operated manually, or automatically operated by a pneumatic or electric motor controlled from a remote temperature or pressure controller in the process stream. Louvers used with constant speed fans do not reduce fan power requirements.

Though use is less common in recent years for air-side control, auto-variable-pitch fans are generally provided with pneumatically operated blade pitch adjustment which may be controlled from a remote sensor. Blade pitch is adjusted to provide the required amount of air flow to maintain the process temperature or pressure at the cooler. The required blade angle decreases as ambient air temperature drops and this conserves fan power. The use of VFDs on fan motors has become one of the most commonly used methods of air-side control in recent exchanger design. VFDs reduce fan speed when less air flow is required and can also conserve fan power. The use of VFDs has been minimal in the past due to the additional cost associated with them. However, decreases in the cost of smaller VFDs (for 50 hp motors and smaller), along with the high cost of maintaining variable pitch fans, have made the use of VFDs more popular in recent practice and design. In fact, most air cooler designs will have VFDs for air side control, despite already having louvers that are required for cooler winterization purposes.

A design consideration which might be required for satisfactory process fluid control is co-current flow. In extreme cases of

high pour point fluids, no amount of air side control would allow satisfactory cooling and prevent freezing. Co-current flow has the coldest air cool the hottest process fluid, while the hottest air cools the coolest process fluid. This is done in order to maintain a high tube wall temperature. This gives a much poorer LMTD, but for highly viscous fluids is often the only way to prevent freezing or unacceptable pressure drops. With air coolers, the most common method of accomplishing co-current flow is to have the inlet nozzle on the bottom of the header with the pass arrangement upwards. This totally reverses the standard design, and may cause a problem with drainage during shut-downs. In addition, air side control is necessary with co-current designs.

WARM AIR RECIRCULATION

Extreme variation in air temperature, such as encountered in northern climates, may require special air recirculation features. These are needed to provide control of process stream temperatures, and to prevent freezing of liquid streams. Warm air recirculation varies from a standard cooler with one reversing fan to a totally enclosed system of automatic louvers and fans. These two widely used systems are termed "internal recirculation" and "external recirculation."

A typical layout for internal recirculation is shown in Fig. 10-6. During low ambient operation, the manual fan continues to force air through the inlet half of the section. The auto-variable fan operates in a reversing mode, and draws hot air from the upper recirculation chamber down through the outlet end of the section. Because of the lower recirculation skirt, the manual fan mixes some of the hot air brought down by the auto-variable fan with cold outside air and the process repeats. The top exhaust louvers are automatically adjusted by a temperature controller sensing the process fluid stream. As the fluid temperature rises, the louvers are opened. During design ambient conditions, the louvers are full open and both fans operate in a standard forced draft mode.

A cooler with internal recirculation is a compromise between no recirculation and fully controlled external recirculation. It is cheaper than full external recirculation, and has less static pressure loss during maximum ambient temperature conditions. A cooler with internal recirculation is easier to erect, and requires less plot area than an external recirculation design. However, this latter design is more costly than a cooler with no recirculation, and cannot provide complete freeze protection. Because there is no control over air intake, and fans alone cannot fully mix air, stratified cold air may contact the section. With the fans off, high wind velocity during low ambient conditions could cause excessive cold air to reach the section.

A typical layout for external recirculation is shown in Fig. 10-7. During low ambient temperature conditions, two-speed motors on low speed, or auto-variable fans at low pitch, are normally used. For this design, the sides of the cooler are closed with manual louvers. Over both ends, a recirculation chamber projects beyond the section headers, and provides a duct for mixing cold outside air with warm recirculated air. As with the internal recirculation design, the top exhaust louvers are controlled by the temperature of the process fluid. However, this design provides for control of the inlet air temperature. As the inlet air louver closes, an internal louver in the end duct opens. These adjustments are determined by a controller which senses air temperature at the fan. Once the system reaches equilibrium, it automatically controls process temperature and prevents excessive cooling. During warm weather, the side manual louvers are opened, while close control is maintained by adjustment of the exhaust louvers.

FIG. 10-5

Typical Construction of Tube Section with Plug and Cover Plate Headers

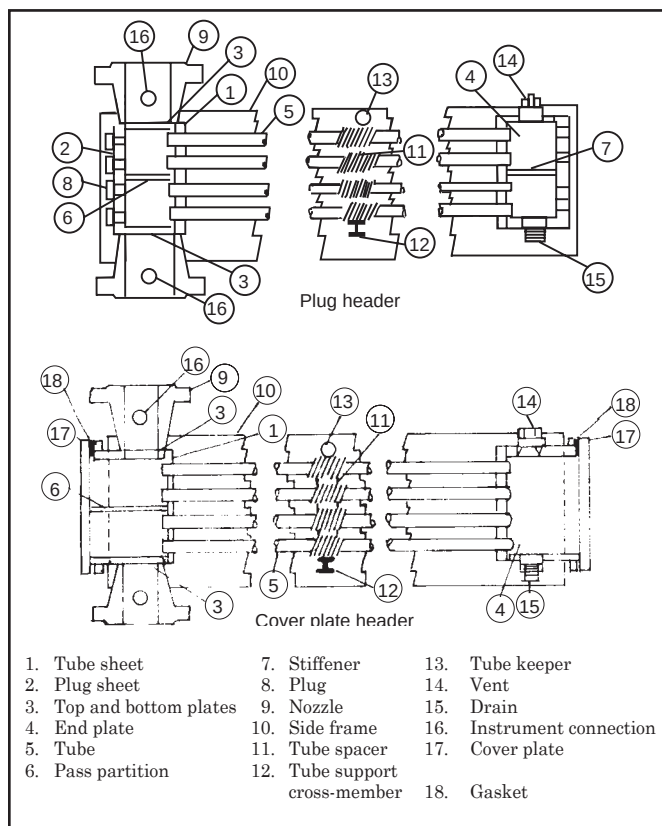


FIG. 10-6
Internal Recirculation Design

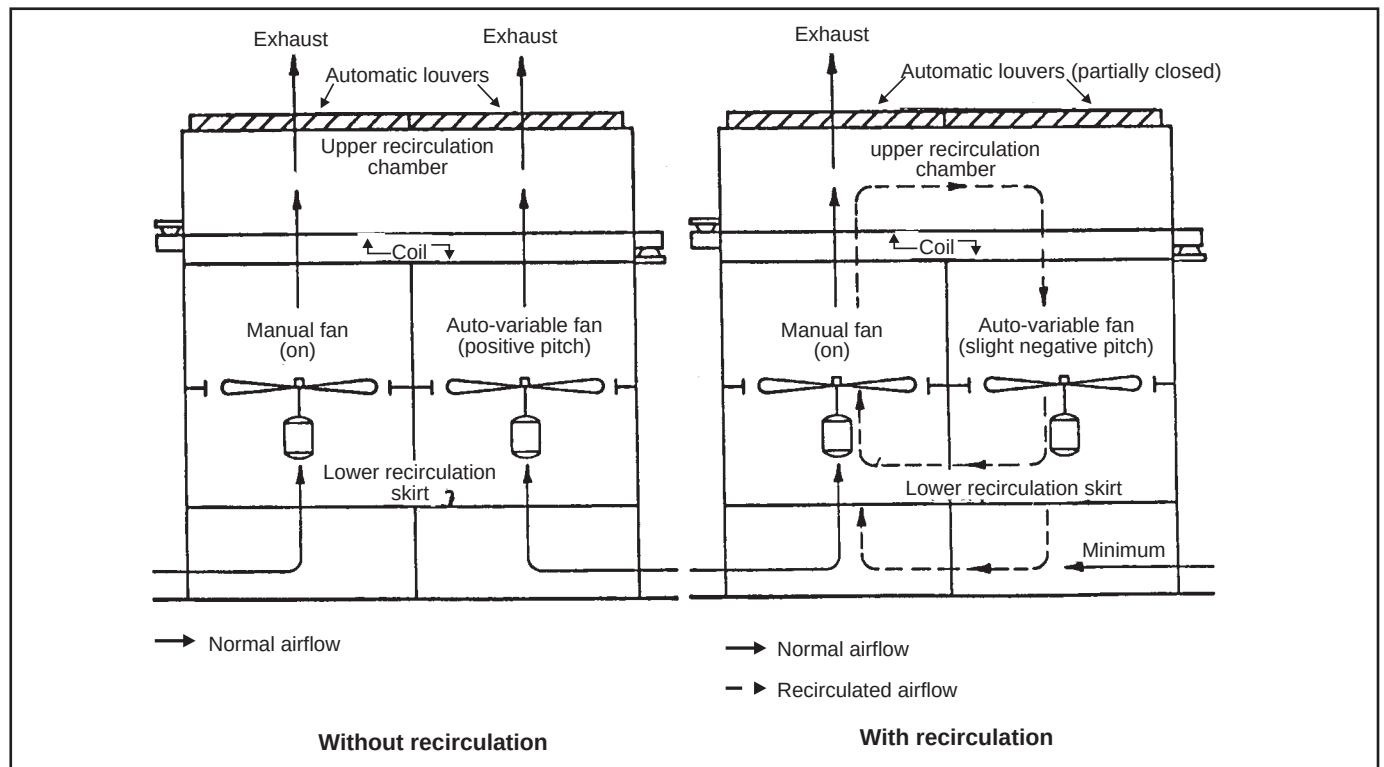
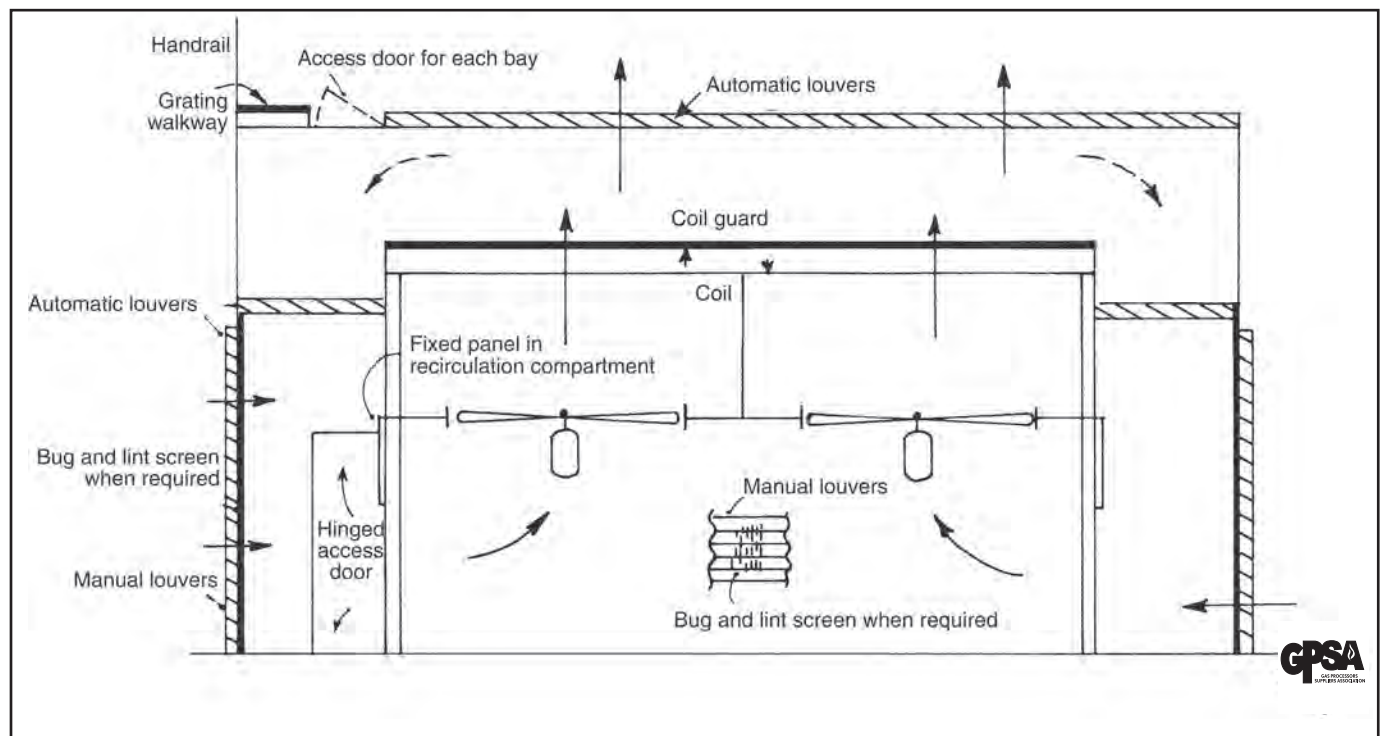


FIG. 10-7
External Recirculation Design



The external recirculation design is preferred for critical control and prevention of freezing. Once operational, it requires little attention. Upon failure of power or air supply, the system closes automatically to prevent freezing. It can be designed to automatically reduce motor energy use when excess cooling is being provided. The main drawback for this type of system is its high cost. Several actuators and control devices are required, along with more steel and louvers. It is usually too large to be shop assembled, and requires more field assembly than an internal system. Because of the need to restrict air intake, this design increases the static pressure, causing greater energy use, and 20-25% larger motors than a standard cooler.

When designing an external recirculation unit, consideration must be given to the plenum depth and duct work to allow air mixing and prevent excessive static pressure loss. The louver intake area should be large enough to keep the air flow below 500 ft/min during maximum design conditions.

AIR EVAPORATIVE COOLERS

Wet/dry type (air evaporative coolers) air coolers may be a good economical choice when a close approach to the ambient temperature is required. In these systems, the designer can take advantage of the difference between the dry bulb and wet bulb temperatures. There are two general types of air evaporative cooler combinations used although other combinations are possible:

Wet air type — In this type, the air is humidified by spraying water into the air stream on the inlet side of the air cooler. The air stream may then pass through a mist eliminator to remove the excess water. The air then passes over the finned tubes at close to its wet-bulb temperature. If the mist eliminator is not used, the spray should be clean, treated water or the tube/fin type and metallurgy should be compatible with the water.

Wet tube type — An air evaporative cooler may be operated in series with an air cooler if there is a large process fluid temperature change with a close approach to the ambient. The process fluid enters a dry finned tube section and then passes into a wet, plain tube section (or appropriate finned tube section). The air is pulled across the wet tube section and then, after dropping out the excess moisture, passes over the dry tube section.

SPECIAL PROBLEMS IN STEAM CONDENSERS

There are often problems with steam condensers which need special attention at the design stage.

Imploding (collapsing bubbles) or knocking can create violent fluid forces which may damage piping or equipment. These forces are created when a subcooled condensate is dumped into a two-phase condensate header, or when live steam passes into subcooled condensate. This problem is avoided by designing the steam system and controls so that steam and subcooled condensate do not meet in the system.

Non-condensable gas stagnation can be a problem in the air cooled steam condenser any time there is more than one tube row per pass. The temperature of the air increases row by row from bottom to top of the air cooled section. The condensing capacity of each row will therefore vary with each tube row in proportion to the ΔT driving force. Since the tubes are connect-

ed to common headers and are subject to the same pressure drop, the vapor flows into the bottom rows from both ends. The non-condensables are trapped within the tube at the point of lowest pressure. The non-condensables continue to accumulate in all but the top rows until they reach the tube outlet. The system becomes stable with the condensate running out of these lower tube rows by gravity. This problem can be eliminated in several ways:

- By assigning only one tube row per pass.
- By connecting the tube rows at the return end with 180° return bends and eliminating the common header.

AIR COOLER LOCATION

Circulation of hot air to the fans of an air cooler can greatly reduce the cooling capacity of an air cooler. Cooler location should take this into consideration.

Single Installations

Avoid locating the air-cooled exchanger too close to buildings or structures in the downwind direction. Hot air venting from the air cooler is carried by the wind, and after striking the obstruction, some of the hot air recycles to the inlet. An induced draft fan with sufficient stack height alleviates this problem, but locating the air cooler away from such obstructions is the best solution.

An air cooler with forced draft fans is always susceptible to air recirculation. If the air cooler is located too close to the ground, causing high inlet velocities relative to the exhaust air velocity leaving the cooler, the hot air recirculation can become very significant. Forced draft coolers are preferably located above pipe lanes relatively high above the ground. Induced draft coolers are less likely to experience recirculation because the exhaust velocities are normally considerably higher than the inlet velocities.

Banks of Coolers

Coolers arranged in a bank should be close together or have air seals between them to prevent recirculation between the units. Mixing of induced draft and forced draft units in close proximity to each other invites recirculation. Avoid placing coolers at different elevations in the same bank.

Avoid placing the bank of coolers downwind from other heat generating equipment.

Since air can only enter on the ends of coolers in a bank, the bank should be located above ground high enough to assure a reasonably low inlet velocity.

The prevailing summer wind direction can have a profound effect on the performance of the coolers. Normally the bank should be oriented such that the wind flows parallel to the long axis of the bank of coolers, and the items with the closest approach to the ambient temperature should be located on the upwind end of the bank.

These generalizations are helpful in locating coolers. The use of Computational Fluid Dynamics to study the effect of wind direction, velocity, obstructions, and heat generating objects should be considered to assure the best location and orientation of air cooled heat exchangers, especially for large installations.