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 dis-

FIG. 10-1

Nomenclature

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

FIG. 10-1

Nomenclature (Cont'd.)

Y = correction factor, psi/ft (see Fig. 10-14)	**Subscripts:**
Δt = temperature change, °F	a = air side
μ = viscosity, cp	b = bare tube surface basis
μ_w = viscosity at average tube wall temperature, cp	s = shell side
φ = viscosity gradient correction	t = tube side
	x = extended tube surface basis
	1 = inlet
	2 = outlet

charged upward at approximately 2½ 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.
- 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:

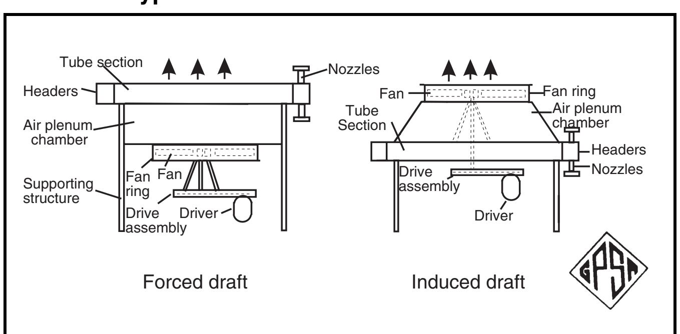
- Slightly lower horsepower since the fan is in cold air. (Horsepower varies directly as the absolute temperature.)
- Better accessibility of mechanical components for maintenance.
- Easily adaptable for warm air recirculation for cold climates.

The disadvantages of forced draft are:

- Poor distribution of air over the section.

FIG. 10-2

Typical Side Elevations of Air Coolers



- 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.

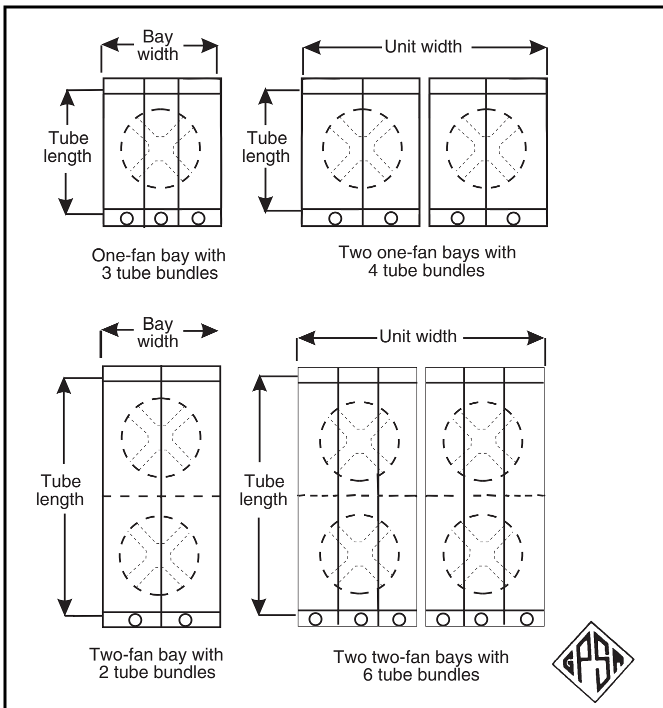
The horizontal section is the most commonly used air cooled section, and generally the most economical. For a fluid with freezing potential, the tubes should be sloped at least ⅛ in. per foot to the outlet header. Since in most cases there will be no problem associated with freezing, and it is more costly 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.

FIG. 10-3

Typical Plan Views of Air Coolers



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 30 bhp and gear drives at higher power. Individual driver size is usually limited to 50 hp.

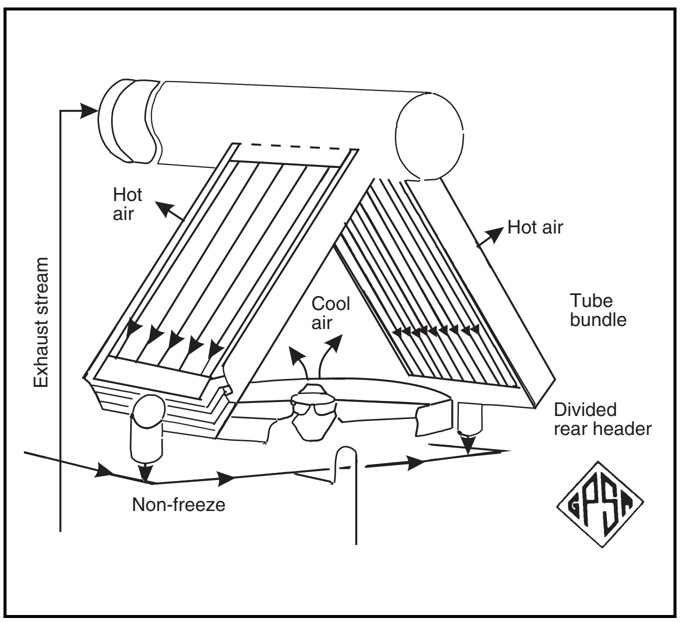
Two fan bays are popular, since this provides a degree of safety against fan or driver failure and also a method of control by fan staging. Fan coverage is the ratio of the projected area of the fan to the face of the section served by the fan. Good practice is to keep this ratio above 0.40 whenever 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.

FIG. 10-4

Angled Section Layout



Use of longer tubes usually results in a less costly design compared to using shorter tubes.

Base tube diameters are ⅝ in. to 1½ in. OD with fins from ½ 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 1⁄16 in. to ¼ 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 4 rows the most typical.

A 1-in. OD tube is the most popular diameter, and the most common fins are ½ in. or ⅝ in. high. The data presented in Fig. 10-11 are for 1 in. OD tubes with ½ in. high fins, 9 fins/in. (designated as ½ x 9) and ⅝ in. high fins, 10 fins/in. (designated as ⅝ x 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.

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

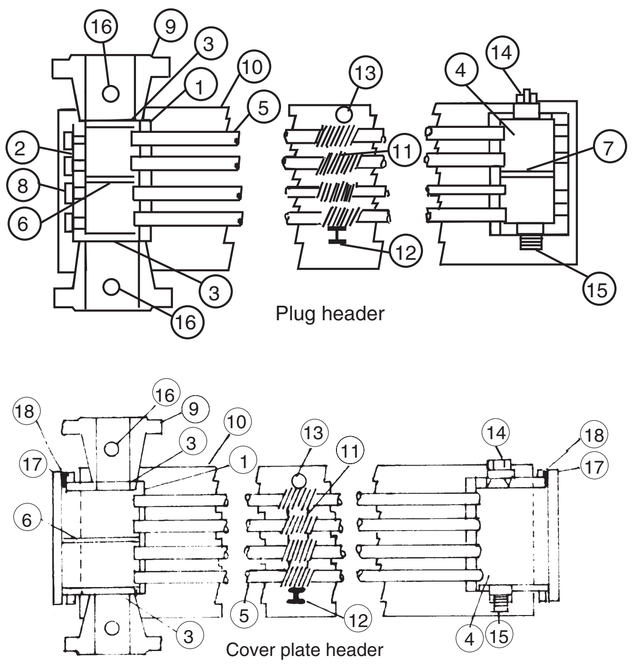
Air-cooled exchangers are sized to operate at warm (summer) air temperatures. Seasonal variation of the air temperature can result in over-cooling which may be undesirable. One way to control the amount of cooling is by varying the amount of air flowing through the tube section. This can be accomplished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

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.

Auto-variable-pitch fans are normally 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. Hydraulic variable speed drives reduce fan

FIG. 10-5

Typical Construction of Tube Section with Plug and Cover Plate Headers



speed when less air flow is required and can also conserve fan power.

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.

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.

FIG. 10-6

Internal Recirculation Design

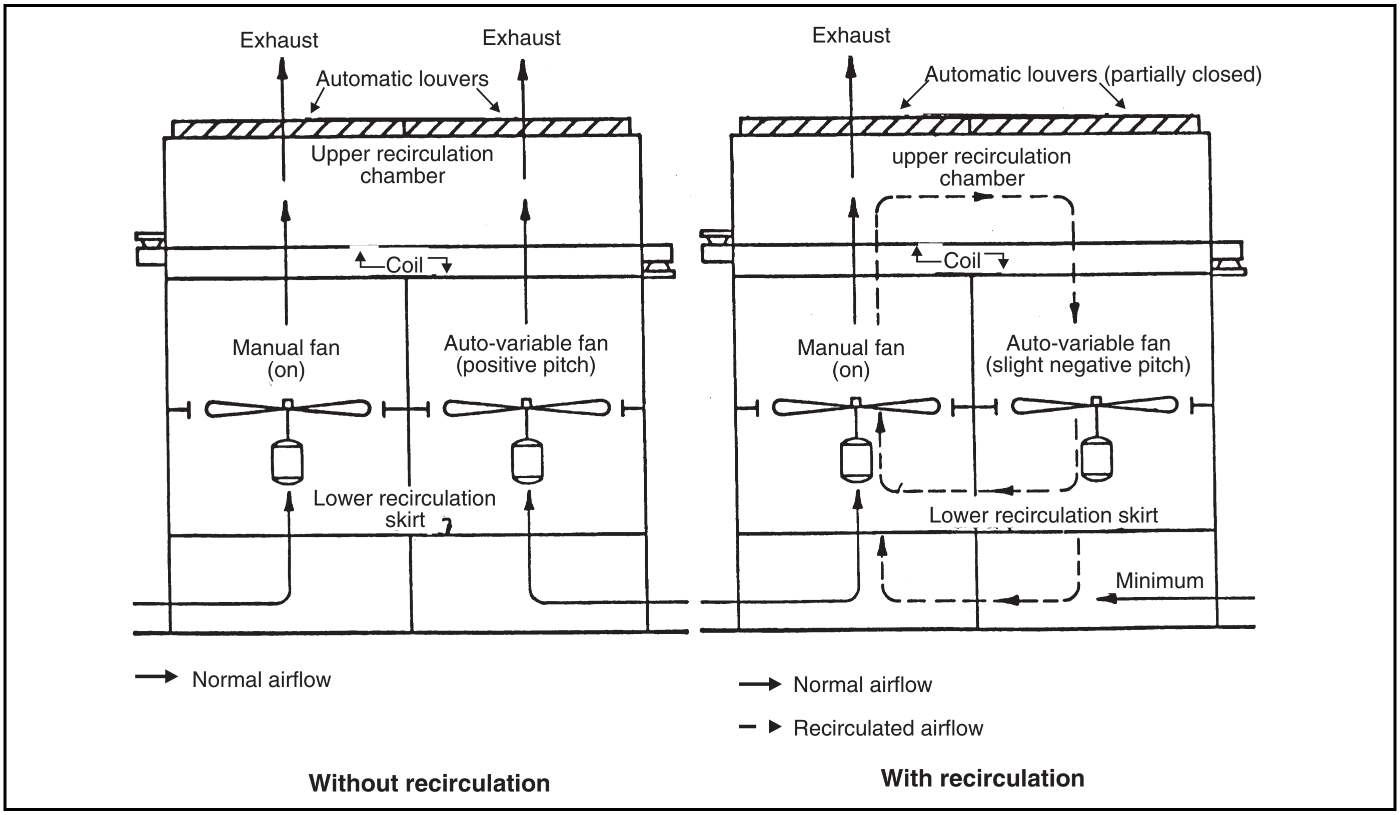
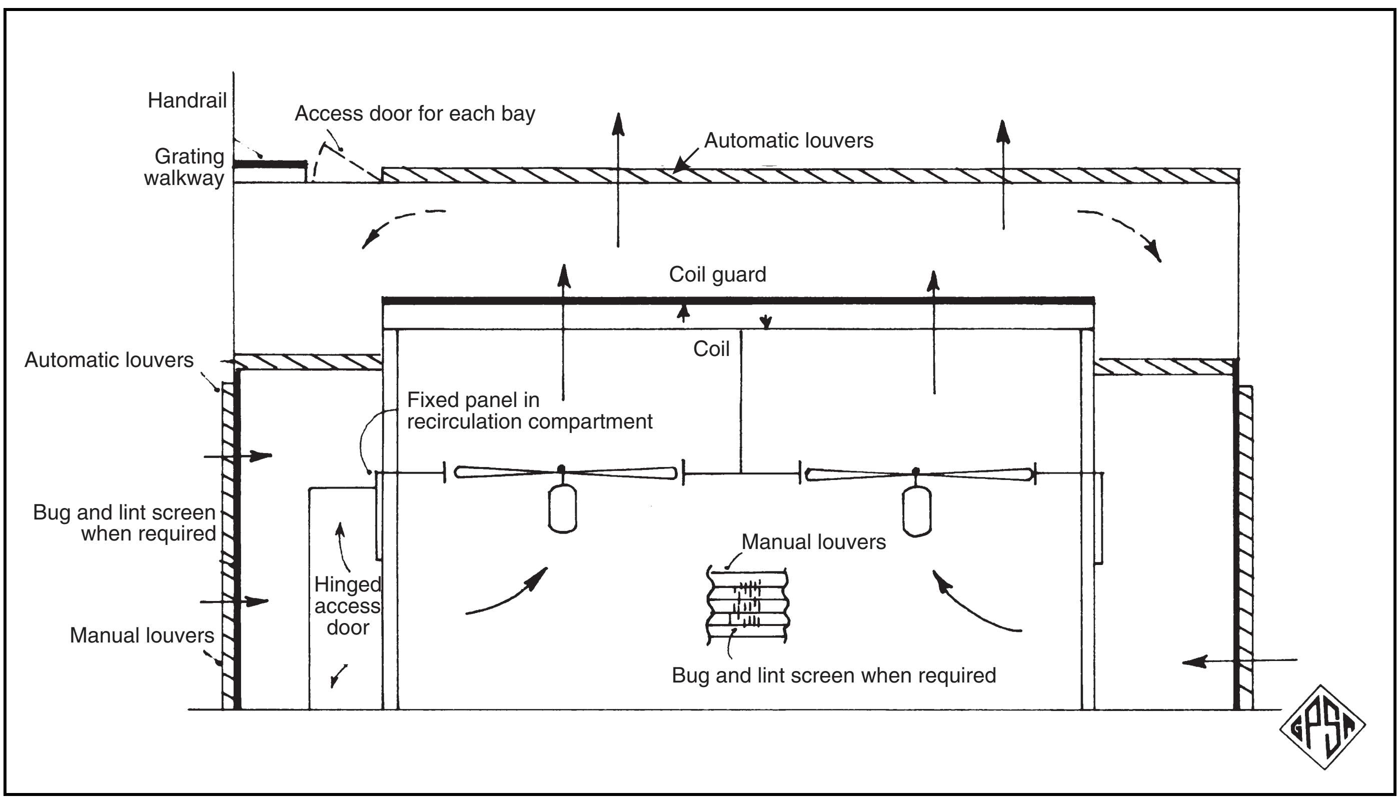


FIG. 10-7

External Recirculation Design



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 connected 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.

MULTIPLE SERVICE DISCUSSION

If different services can be placed in the same plot area without excessive piping runs, it is usually less expensive to combine them on one structure, with each service having a separate section, but sharing the same fan and motors. Separate louvers may be placed on each service to allow independent control. The cost and space savings makes this method common practice in the air cooler industry.

In designing multiple service coolers, the service with the most critical pressure drop should be calculated first. This is because the pressure drop on the critical item might restrict the maximum tube length that the other services could tolerate. The burden of forcing more than one service into a single tube length increases the possibility of design errors. Several trial calculations may be needed to obtain an efficient design.

After all service plot areas have been estimated, combine them into a unit having a ratio of 2 or 3 to 1 in length to width (assuming a two fan cooler). After assuming a tube length, calculate the most critical service for pressure drop using the assumed number and length of tubes and a single pass. If the drop is acceptable or very close, calculate the critical service completely. Once a design for the most critical service has been completed, follow the same procedure with the next most critical service. After the second or subsequent services have been rated, it is often necessary to lengthen or shorten the tubes or change the overall arrangement. If tubes need to be added for pressure drop reductions in already oversurfaced sections, it might be more cost effective to add a row(s) rather than widen the entire unit. The fan and motor calculations are the same as for a single service unit, except that the quantity of air used must be the sum of air required by all services.

CONDENSING DISCUSSION

The example given covers cooling problems and would work with straight line condensing problems that have the approximate range of dew point to bubble point of the fluid. Where de-superheating or subcooling or where disproportionate amounts of condensing occur at certain temperatures, as with steam and non-condensables, calculations for air coolers should be done by "zones." A heat release curve developed from enthalpy data will show the quantity of heat to be dissipated between various temperatures. The zones to be calculated should be straight line zones; that is, from the inlet temperature of a zone to its outlet, the heat load per degree temperature is the same.

After the zones are determined, an approximate rate must be found for each zone. Do this by taking rates from vapor cooling, condensing, and liquid cooling, then average these based on the percent of heat load for that phase within the zone. Next, calculate the LMTD of each zone. Begin with the outlet zone using the final design outlet temperature and the inlet temperature of that zone. Continue to calculate the zone as if it were a cooler, except that only one pass and one or two rows should be assumed, depending on the percentage of heat load in that zone. In calculating the pressure drop, average conditions may be used for estimating.

If the calculations for zone one (or later a succeeding zone) show a large number of short tubes with one pass, as is usually the case with steam and non-condensables, recalculate the zone with multiple rows (usually four) and short tubes having one pass that uses only a percentage of the total pressure drop allowed. The total cooler will be calculated as if each zone were a cooler connected in series to the next one, except that only tube pressure drops should be calculated for the middle zones. Thus, each zone must have the same number of tubes and true ambient must be used in calculating the LMTD. Only the tube length may vary, with odd lengths for a zone acceptable as long as overall length is rounded to a standard tube length.

If the calculations for zone one (and succeeding zones) fit well into a longer tube length, the LMTD must be weighted. After the outlet zone has been calculated, calculate zone two using the inlet temperature for it and its outlet temperature, which is the inlet temperature of zone one. The "ambient" used to find the zone two LMTD will be the design ambient plus the air rise from zone one. Continue in this manner, always using the previous zone's outlet air temperature in calculating the current zone's LMTD. After the cooler size and configuration have been determined, the fan and motor calculations will be made in the normal manner.

The ultimate pressure drop is the sum of the drops for each zone or approximately the sum of the drop for each phase using the tube length and pass arrangement for each phase. An estimated overall tube side coefficient may be calculated by estimating the coefficient for each phase. Then take a weighted average based on the percentage of heat load for each phase. The total LMTD must be the weighted average of the calculated zone LMTDs.

THERMAL DESIGN

The basic equation to be satisfied is the same as given in Section 9, Heat Exchangers:

$$Q = UA\ CMTD \qquad \textbf{Eq 10-1}$$

Normally Q is known, U and CMTD are calculated, and the equation is solved for A. The ambient air temperature to be used will either be known from available plant data or can be selected from the summer dry bulb temperature data given in Section 11, Cooling Towers. The design ambient air temperature is usually considered to be the dry bulb temperature that is exceeded less than 5 percent of the time in the area where the installation is required.

A complication arises in calculating the LMTD because the air quantity is a variable, and therefore the air outlet temperature is not known. The procedure given here starts with a step for approximating the air-temperature rise. After the air-outlet temperature has been determined, the corrected LMTD is calculated in the manner described in the shell and tube section, except that MTD correction factors to be used are from Figs. 10-8 and 10-9 which have been developed for the crossflow situation existing in air-cooled exchangers.

Fig. 10-8 is for one tube pass. It is also used for multiple tube passes if passes are side by side. Fig. 10-9 is for two tube passes and is used if the tube passes are over and under each other. A MTD correction factor of 1.0 is used for four or more passes, if passes are over and under each other. A correction factor of 1.0 may be used as an approximation for three passes, although the factor will be slightly lower than 1.0 in some cases.

The procedure for the thermal design of an air cooler consists of assuming a selection and then proving it to be correct. The typical overall heat transfer coefficients given in Fig. 10-10 are used to approximate the heat transfer area required. The heat transfer area is converted to a bundle face area using Fig. 10-11 which lists the amount of extended surface available per square foot of bundle area for two specific fin tubes on two different tube pitches for 3, 4, 5, and 6 rows. After assuming a tube length, Fig. 10-11 is also used to ascertain the number of tubes. Both the tube side and air side mass velocities are now determinable.

The tube-side film coefficient is calculated from Figs. 10-12 and 10-13. Fig. 10-17 gives the air-side film coefficient based on outside extended surface. Since all resistances must be based on the same surface, it is necessary to multiply the reciprocal of the tube-side film coefficient and tube-side fouling factor by the ratio of the outside surface to inside surface. This results in an overall transfer rate based on extended surface, designated as U_x. The equation for overall heat transfer rate is:

$$\frac{1}{U_x} = \left(\frac{1}{h_t}\right)\left(\frac{A_x}{A_i}\right) + r_{dt}\left(\frac{A_x}{A_i}\right) + r_{mx} + \frac{1}{h_a} \qquad \textbf{Eq 10-2}$$

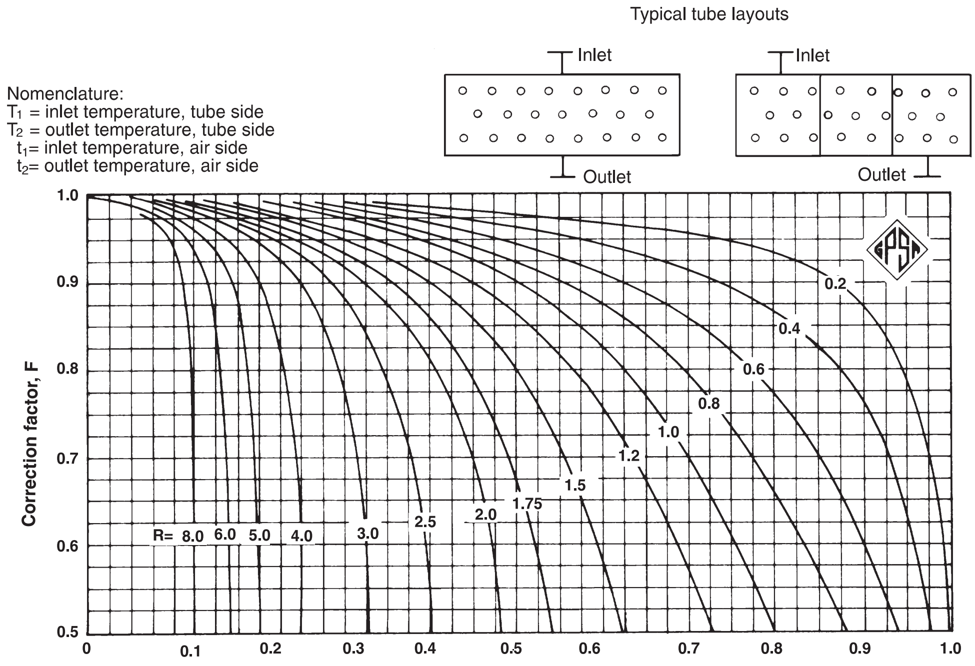
The basic equation will then yield a heat transfer area in extended surface, A_x, and becomes:

$$Q = (U_x)\,(A_x)\ CMTD$$

Either method is valid and each is used extensively by thermal design engineers. Fig. 10-10 gives typical overall heat

FIG. 10-8

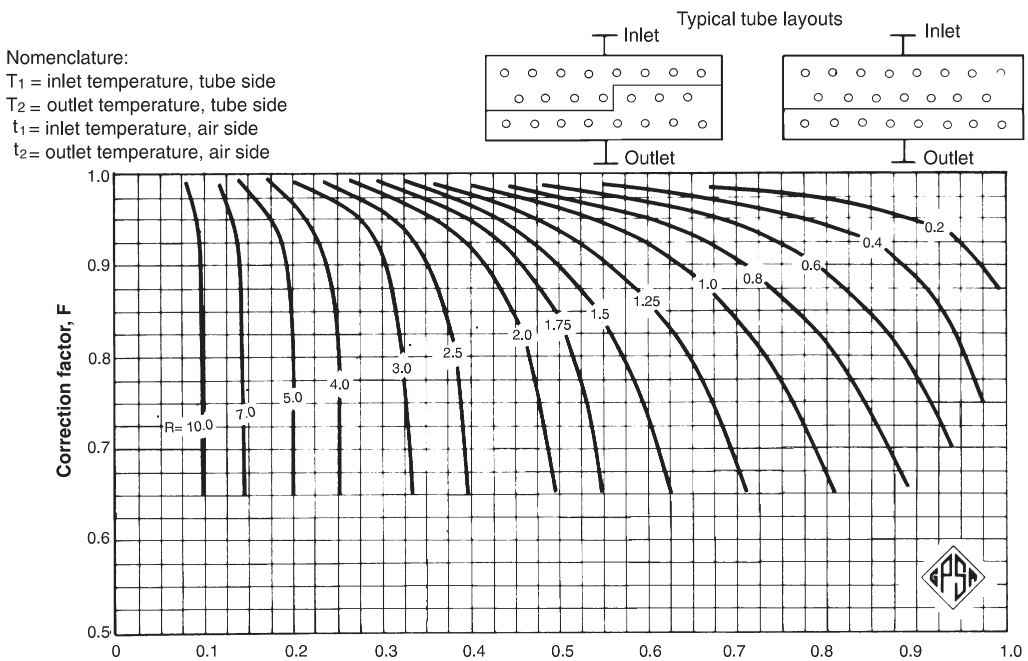
MTD Correction Factors (1 Pass – Cross Flow, Both Fluids Unmixed)



$$R = \frac{T_1 - T_2}{t_2 - t_1} \qquad P = \frac{t_2 - t_1}{T_1 - t_1}$$

FIG. 10-9

MTD Correction Factors (2 Pass – Cross Flow, Both Fluids Unmixed)



$$R = \frac{T_1 - T_2}{t_2 - t_1} \qquad P = \frac{t_2 - t_1}{T_1 - t_1}$$

FIG. 10-10

Typical Overall Heat-Transfer Coefficients for Air Coolers

Service	1 in. Fintube			
	½ in. by 9		⅝ in. by 10	
	U_b	U_x	U_b	U_x
1. Water & water solutions	(See note below)			
Engine jacket water (r_d = 0.001)	110	7.5	130	6.1
Process water (r_d = 0.002)	95	6.5	110	5.2
50-50 ethylene glycol-water (r_d = 0.001)	90	6.2	105	4.9
50-50 ethylene glycol-water (r_d = 0.002)	80	5.5	95	4.4
2. Hydrocarbon liquid coolers				
Viscosity, cp, at avg. temp.	U_b	U_x	U_b	U_x
0.2	85	5.9	100	4.7
0.5	75	5.2	90	4.2
1.0	65	4.5	75	3.5
2.5	45	3.1	55	2.6
4.0	30	2.1	35	1.6
6.0	20	1.4	25	1.2
10.0	10	0.7	13	0.6
3. Hydrocarbon gas coolers				
Pressure, psig	U_b	U_x	U_b	U_x
50	30	2.1	35	1.6
100	35	2.4	40	1.9
300	45	3.1	55	2.6
500	55	3.8	65	3.0
750	65	4.5	75	3.5
1000	75	5.2	90	4.2
4. Air and flue-gas coolers Use one-half of value given for hydrocarbon gas coolers.				
5. Steam Condensers (Atmospheric pressure & above)	U_b	U_x	U_b	U_x
Pure Steam (r_d = 0.0005)	125	8.6	145	6.8
Steam with non-condensibles	60	4.1	70	3.3
6. HC condensers				
Condensing* Range, °F	U_b	U_x	U_b	U_x
0° range	85	5.9	100	4.7
10° range	80	5.5	95	4.4
25° range	75	5.2	90	4.2
60° range	65	4.5	75	3.5
100° & over range	60	4.1	70	3.3
7. Other condensers	U_b	U_x	U_b	U_x
Ammonia	110	7.6	130	6.1
Freon 12	65	4.5	75	3.5

Notes: U_b is overall rate based on bare tube area, and U_x is overall rate based on extended surface.

Based on approximate air face mass velocities between 2600 and 2800 lb/(hr·sq ft of face area).

*Condensing range = hydrocarbon inlet temperature to condensing zone minus hydrocarbon outlet temperature from condensing zone.

transfer coefficients based on both extended surface and outside bare surface, so either method may be used. The extended surface method has been selected for use in the example which follows. The air-film coefficient in Fig. 10-17 and the air static pressure drop in Fig. 10-18 are only for 1 in. OD tubes with ⅝ in. high fins, 10 fins per inch on 2¼ in. triangular pitch. Refer to Bibliography Nos. 2, 3, and 5 for information on other fin configurations and spacings.

The minimum fan area is calculated in Step 16 using the bundle face area, number of fans, and a minimum fan coverage of 0.40. The calculated area is then converted to a diameter and rounded up to the next available fan size. The air-side static pressure is calculated from Fig. 10-18 and the fan total pressure is estimated using gross fan area in Step 20. Finally, fan horsepower is calculated in Step 21 assuming a fan efficiency of 70%, and driver horsepower is estimated by assuming a 92%-efficient speed reducer.

Example 10-1 — Procedure for estimating transfer surface, plot area, and horsepower

Required data for hot fluid

Name and phase: 48°API hydrocarbon liquid

Physical properties at avg temp = 200°F

C_p = 0.55 Btu/(lb • °F)

μ = 0.51 cp

k = 0.0766 Btu/[(hr • sq ft • °F)/ft]

(From this Data Book Section 23)

Heat load: Q = 15,000,000 Btu/hr

Flow quantity: W_t = 273,000 lb/hr

Temperature in: T_1 = 250°F

Temperature out: T_2 = 150°F

Fouling factor r_{dt} = 0.001 (hr • sq ft • °F)/Btu

Allowable pressure drop: ΔP_t = 5 psi

Required data for air

Ambient temperature: t_1 = 100°F

Elevation: Sea level (see Fig. 10-16 for altitude correction)

C_{Pair} = 0.24 Btu/(lb • °F)

Basic assumptions

Type: Forced draft, 2 fans

Fintube: 1 in. OD with ⅝ in. high fins

Tube pitch: 2 ½ in. triangular (Δ)

Bundle layout: 3 tube passes, 4 rows of tubes, 30 ft long tubes

First trial

1. Pick approximate overall transfer coefficient from Fig. 10-10. U_x = 4.2

FIG. 10-11

Fintube Data for 1-in. OD Tubes

Fin Height by Fins/inch	**½ in. by 9**		**⅝ in. by 10**		
APF, sq ft/ft	3.80		5.58		
AR, sq ft/sq ft	14.5		21.4		
Tube Pitch	2 in. Δ	2¼ in. Δ	2¼ in. Δ	2⅜ in. Δ	2½ in. Δ
APSF (3 rows)	68.4	60.6	89.1	84.8	80.4
(4 rows)	91.2	80.8	118.8	113.0	107.2
(5 rows)	114.0	101.0	148.5	141.3	134.0
(6 rows)	136.8	121.2	178.2	169.6	160.8

Notes: APF is total external area/ft of fintube in sq ft/ft. AR is the area ratio of fintube compared to the exterior area of 1 in. OD bare tube which has 0.262 sq ft/ft. APSF is the external area in sq ft/sq ft of bundle face area.

2. Calculate approximate air temperature rise

$$\Delta t_a = \left(\frac{U_x + 1}{10}\right)\left(\frac{T_1 + T_2}{2} - t_1\right)$$

$$\Delta t_a = \left(\frac{4.2 + 1.0}{10}\right)\left(\frac{250 + 150}{2} - 100\right) = 52°F$$

3. Calculate CMTD

$$\begin{array}{lccc} \text{Hydocarbon} & 250 & \rightarrow & 150 \\ \text{Air} & \underline{152} & \leftarrow & \underline{100} \\ & 98 & & 50 \end{array}$$

LMTD = 71.3°F (see Fig. 9-3)

CMTD = (71.3) (1.00) = 71.3°F
(3 tube passes assumed)

4. Calculate required surface

$$A_x = \frac{Q}{(U_x)\ (CMTD)}$$

$$A_x = \frac{15{,}000{,}000}{(4.2)\ (71.3)} = 50{,}090 \text{ sq ft}$$

5. Calculate face area using APSF factor from Fig. 10-11

$$F_a = \frac{A_x}{APSF}$$

$$F_a = \frac{50{,}090}{107.2} = 467 \text{ sq ft (4 rows assumed)}$$

6. Calculate unit width with assumed tube length

$$\text{Width} = \frac{F_a}{L}$$

$$\text{Width} = \frac{467}{30} = 15.57 \text{ ft}$$

For simplification round this answer to 15.5, thus F_a = 465 (30-ft-long tubes assumed)

7. Calculate number of tubes using APF factor from Fig. 10-11

$$N_t = \frac{A_x}{(APF)\ (L)}$$

8. Calculate tube-side mass velocity from assumed number of passes and reading A_t from Fig. 9-25 for a 1 in. OD x 16 BWG tube

A_t = 0.5945 sq in.

$$G_t = \frac{(144)\ (W_t)\ (N_p)}{(3600)\ (N_t)\ (A_t)}$$

$$G_t = \frac{(0.04)\ (273{,}000)\ (3)}{(299)\ (0.5945)} = 184 \text{ lb/(ft}^2 \bullet \text{sec)}$$

9. Calculate modified Reynolds number

$$N_R = \frac{(D_i)\ (G_t)}{\mu} = \frac{(0.87)\ (184)}{0.51} = 314$$

10. Calculate tube-side pressure drop using equation from Fig. 10-14 and from Fig. 10-15

$$\Delta P_t = \frac{fYLN_p}{\phi} + BN_p$$

$$\Delta P_t = \frac{(0.0024)\ (14.5)\ (30)\ (3)}{0.96} + (0.25)\ (3) = 4.0 \text{ psi}$$

(ϕ is a difficult function to calculate rigorously, see Fig. 10-19)

11. Calculate tube-side film coefficient using equation from Fig. 10-13 and

$$k\left(\frac{C_p\mu}{k}\right)^{1/3} \text{ from Fig. 10-12}$$

$$h_t = \frac{Jk\left(\frac{C_p\mu}{k}\right)^{1/3}\phi}{D_i} = \frac{(1900)\ (0.12)\ (0.96)}{0.87} = 252$$

12. Calculate air quantity

$$W_a = \frac{Q}{(0.24)\ (\Delta t_a)}$$

$$W_a = \frac{15{,}000{,}000}{(0.24)\ (52)} = 1{,}200{,}000 \text{ lb/hr}$$

13. Calculate air face mass velocity

$$G_a = \frac{W_a}{F_a} = \text{lb/(hr} \bullet \text{sq ft of face area)}$$

$$G_a = \frac{1{,}200{,}000}{465} = 2{,}581$$

14. Read air-side film coefficient from Fig. 10-17

h_a = 8.5

15. Calculate overall transfer coefficient

FIG. 10-12

Physical Property Factor for Hydrocarbon Liquids

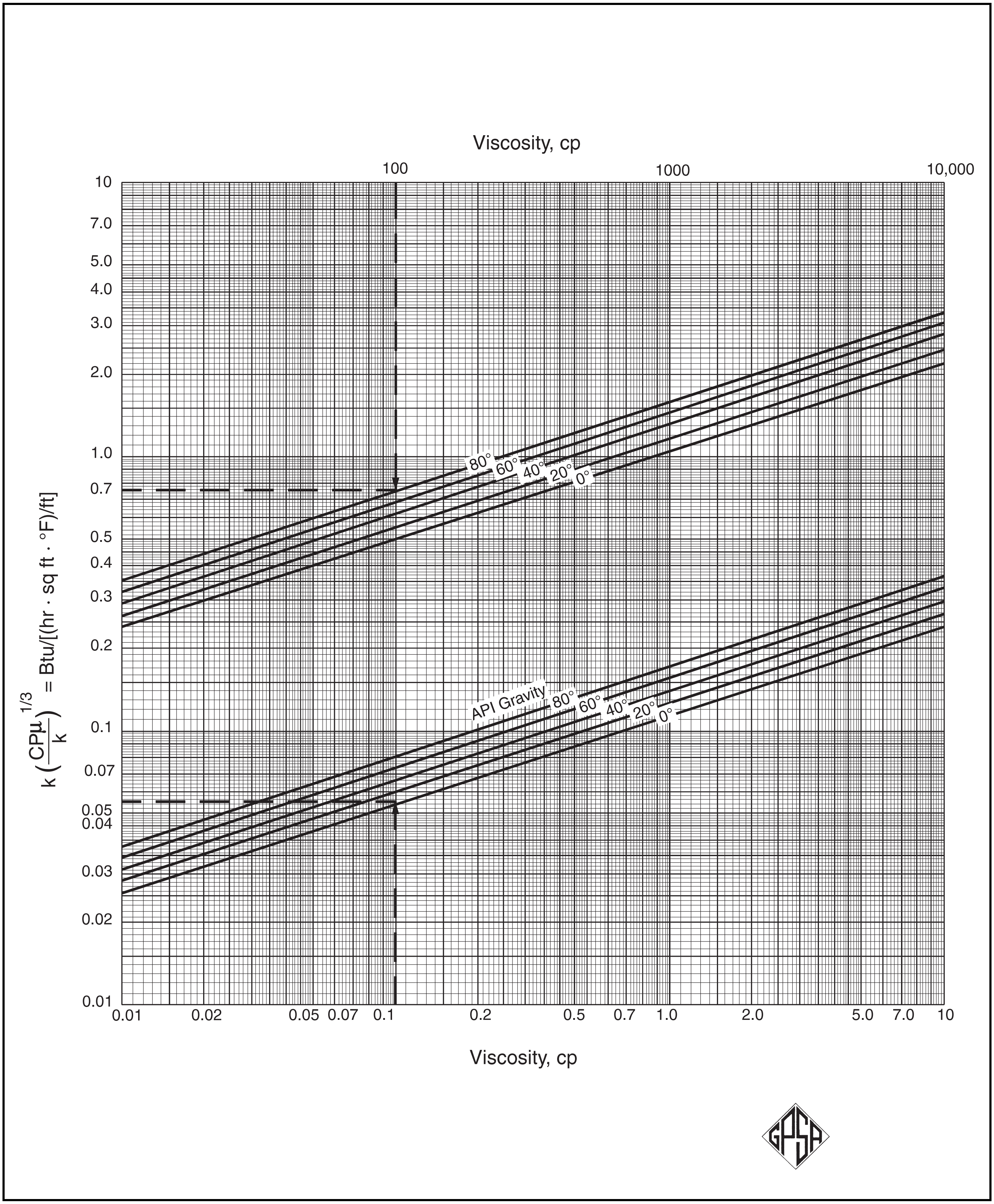


FIG. 10-13

J Factor Correlation to Calculate Inside Film Coefficient, h_t

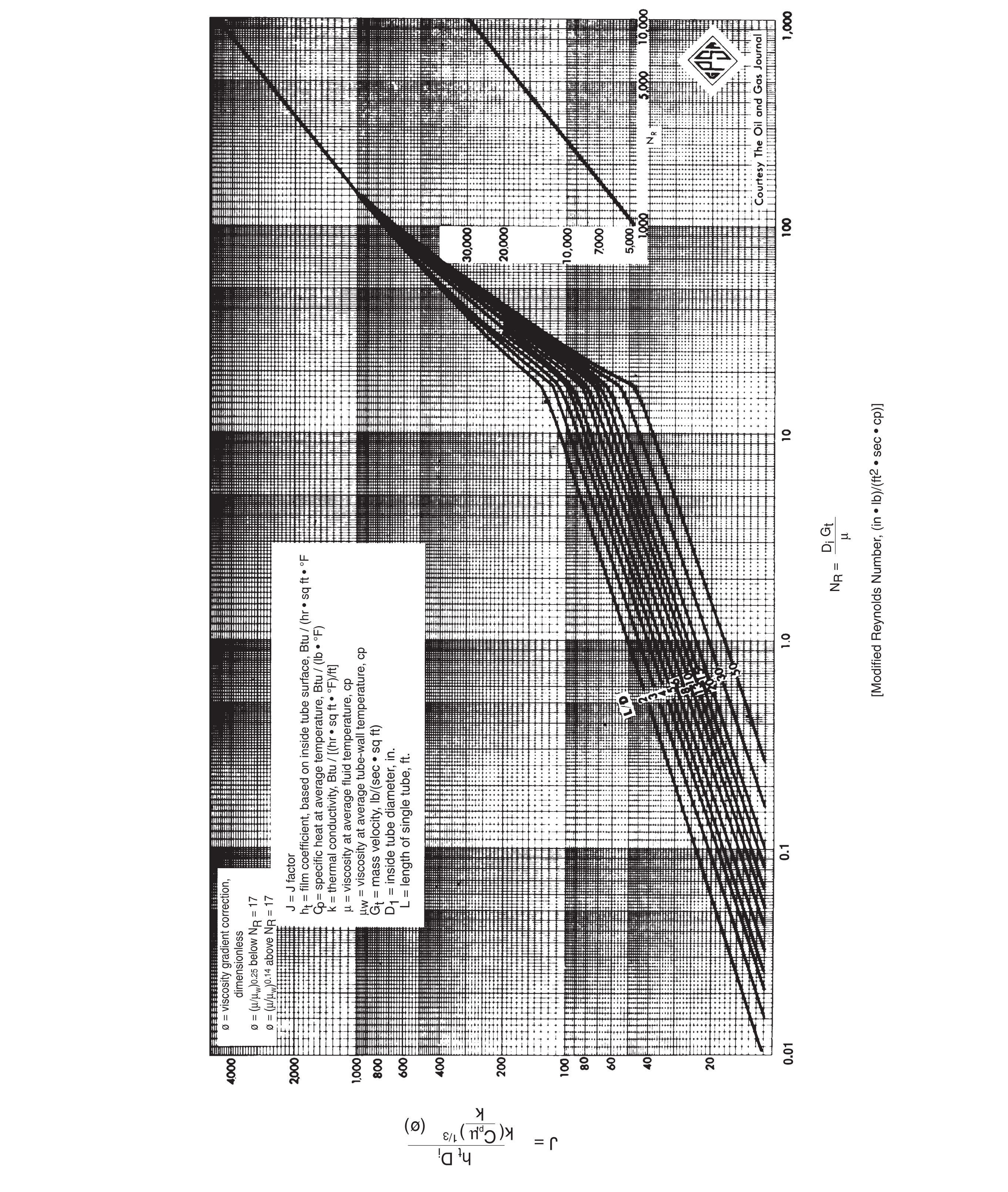


FIG. 10-14

Pressure Drop for Fluids Flowing Inside Tubes

$$\frac{A_x}{A_i} = \frac{(AR)\,(D_o)}{D_i}$$

$$\frac{A_x}{A_i} = \frac{(21.4)\,(1.0)}{0.87} = 24.6$$

$$\frac{1}{U_x} = \left(\frac{1}{h_t}\right)\left(\frac{A_x}{A_i}\right) + r_{dt}\left(\frac{A_x}{A_i}\right) + r_{mx} + \frac{1}{h_a}$$

$$\frac{1}{U_x} = \left(\frac{1}{252}\right)(24.6) + (0.001)\,(24.6) + \frac{1}{8.5}$$

$$U_x = 4.17$$

(r_{mx} is omitted from calculations, since metal resistance is small compared to other resistances)

Second and subsequent trials. If U_x calculated in Step 15 is equal or slightly greater than U_x assumed in Step 1, and calculated pressure drop in Step 9 is within allowable pressure drop, the solution is acceptable. Proceed to Step 16. Otherwise, repeat Steps 1-15 as follows:

1. Assume new U_x between value originally assumed in Step 1 and value calculated in Step 15.
2. Adjust Δt_a by increasing Δt_a if calculated U_x is higher than assumed U_x, or decreasing Δt_a if calculated U_x is lower than assumed U_x.

3.-15. Recalculate values in Steps 3-15 changing assumed number of passes in Steps 3 and 8, and tube length in Step 6, if necessary to obtain a pressure drop as calculated in Step 9 as high as possible without exceeding the allowable.

16. Calculate minimum fan area.

$$\text{Fan area/fan} = \text{FAPF} = \frac{(0.40)\,(F_a)}{(N_f)}$$

$$\text{FAPF} = \frac{(0.40)\,(465)}{2} = 93\text{ ft}^2\text{ (2 fans assumed)}$$

17. Fan diameter $= [4\,(\text{FAPF})/\pi]^{0.5} = [4\,(93)/3.1416]^{0.5}$

$= 11$ ft (rounded up)

18. Calculate air static pressure drop using F_p from Fig. 10-18 and D_R at avg air temp from Fig. 10-16.

$$T_a\text{, avg} = \frac{100°F + 152°F}{2} = 126°F$$

$$\Delta P_a = \frac{(F_p)\,(N)}{(D_R)}$$

$$\Delta P_a = \frac{(0.10)\,(4)}{0.90} = 0.44\text{ inches of water}$$

19. Calculate actual air volume using D_R of air at fan inlet.

$$t_1 = 100°F$$

$$\text{ACFM} = \frac{W_a}{(D_R)\,(60)\,(0.0749)}$$

$$\text{ACFM} = \frac{1{,}200{,}000}{(0.94)\,(60)\,(0.0749)} = 284{,}000\text{ Total}$$

or 142,000 / Fan

20. Approximate fan total pressure using D_R of air at fan and fan area.

$$\text{PF} = \Delta P_a + \left[\frac{\text{ACFM}}{4005\left(\frac{\pi D^2}{4}\right)}\right]^2 (D_R)$$

$$\text{Where:}\quad 4005 = \sqrt{\frac{2\,g\,\rho_w\,(3600)}{\rho_a \bullet 12}}\text{ at }70°F$$

$$\text{PF} = 0.44 + \left(\frac{142{,}000}{(4005)\,(0.785)\,(11^2)}\right)^2 (0.94)$$

$= 0.57$ inches of water

21. Approximate brake horsepower per fan, using 70% fan efficiency.

$$\text{bhp} = \frac{(\text{ACFM/fan})\,(\text{PF})}{(6356)\,(0.70)}$$

Where the conversion factor

$$6356 = \left(\frac{33{,}000\text{ ft–lb}}{\text{min} \bullet \text{hp}}\right)\left(\frac{12\text{ in.}}{\text{ft}}\right)\left(\frac{\text{ft}^3}{62.3\text{ lb}}\right)$$

Note: 62.3 is the weight of one cubic foot of water at 60°F.

$$\text{bhp} = \frac{(142{,}000)\,(0.57)}{(6356)\,(0.70)} = 18.2$$

Actual fan motor needed for 92% efficient speed reducer is 18.2/0.92 = 19.8 hp. For this application, 20 hp drivers would probably be selected.

Solution:

(15.5 ft) (30 ft) = 465 sq ft

(465 sq ft) (APSF) = extended surface area

(465) (107.2) = 49,848 sq ft

Therefore, one unit having 49,848 sq ft of extended surface, two 11 ft diameter fans, and two 20 hp fan drivers, is required.

Fig. 10-20 has been included to aid the air cooler designer in choosing the proper pressure for the air density calculation at elevations higher than sea level.

MAINTENANCE AND INSPECTION

Attention to the design of the air cooler, and the choice of materials, is essential to provide low maintenance operation. Major factors to be considered are atmospheric corrosion, climatic conditions, and temperature cycling of fluid being cooled.

Scheduled preventive maintenance and inspection is the key to trouble-free air cooler operation. A check of all fans for vibration should be made regularly. At the first sign of undue vibration on a unit, the unit should be shut down at the earliest opportunity for thorough examination of all moving parts. A semi-annual inspection and maintenance program should:

- Check and replace worn or cracked belts.
- Inspect fan blades for deflection and for cracks near hubs.
- Grease all bearings.
- Change oil in gear drives.

FIG. 10-15

Friction Factor for Fluids Flowing Inside Tubes

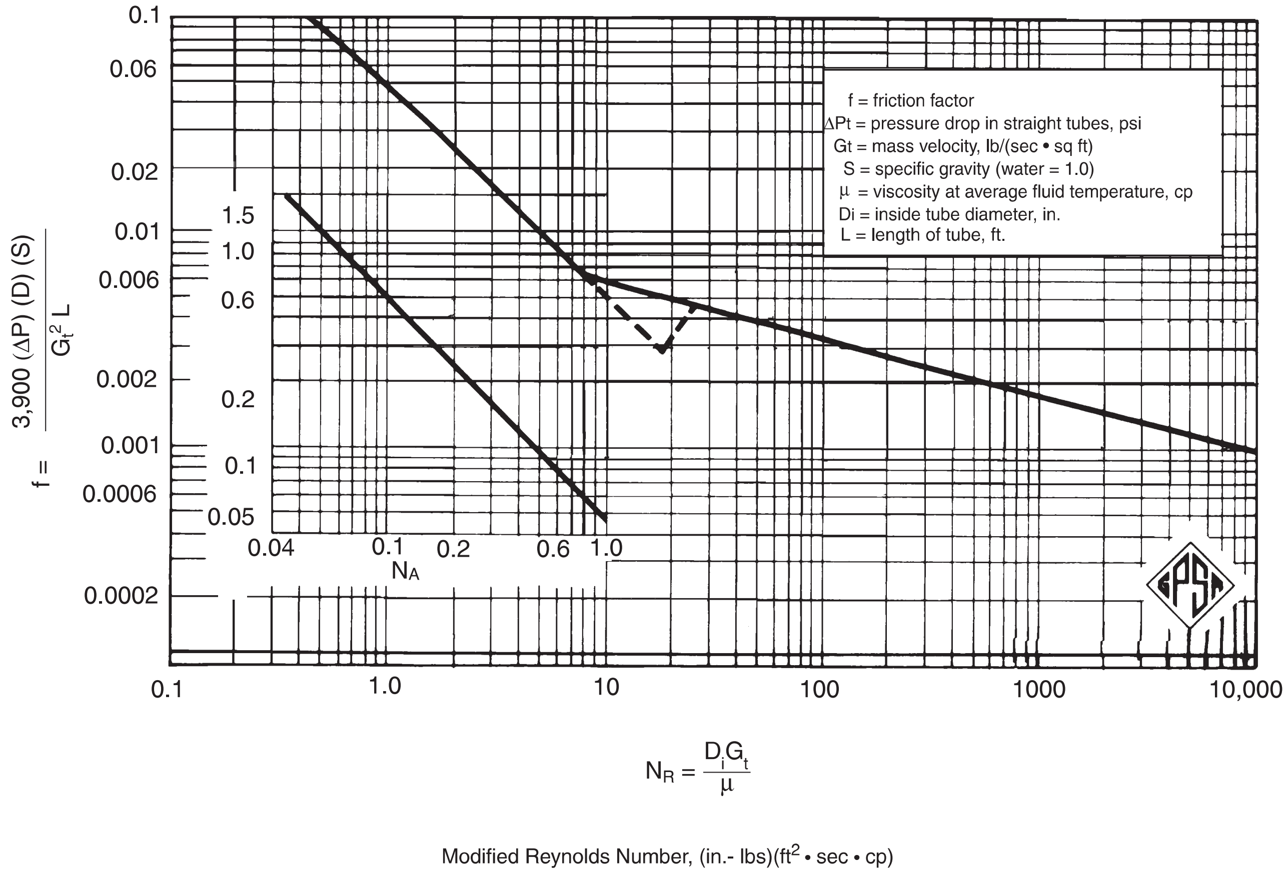


FIG. 10-16

Air-Density Ratio Chart

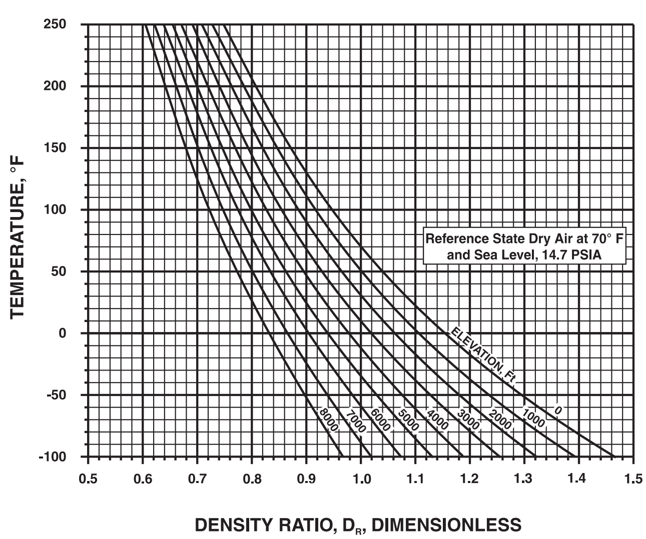
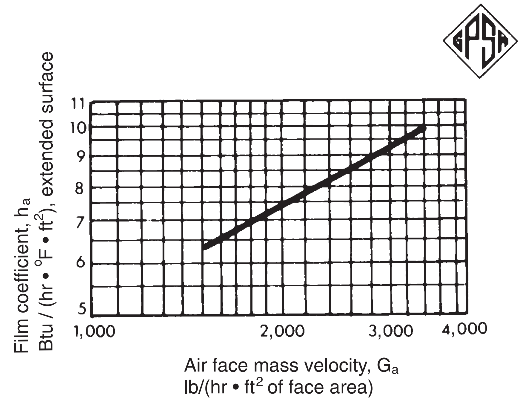


FIG. 10-17

Air Film Coefficient



Note: Figs 10-17 and 10-18 apply only to 1 in. OD, 5/8 in. high, 10 fpi, 2 1/2 in. Δ pitch fintube banks.

- Check the inside of tube section for accumulation of grease, dirt, bugs, leaves, etc., and schedule cleaning before tubes become packed with such debris.

NOISE CONSIDERATIONS

Fan noise can be elusive requiring sophisticated equipment to measure accurately. Fan noise control must begin at the air-cooled heat exchanger design stage. Noise control, as an after thought, can result in a very costly fan and drive component retrofit, and possible addition of heat transfer surface or acoustic barriers. Acoustic barriers may increase the pressure drop the fan must overcome; hence, bigger fans and more horsepower will be required.

FIG. 10-18

Air Static-Pressure Drop

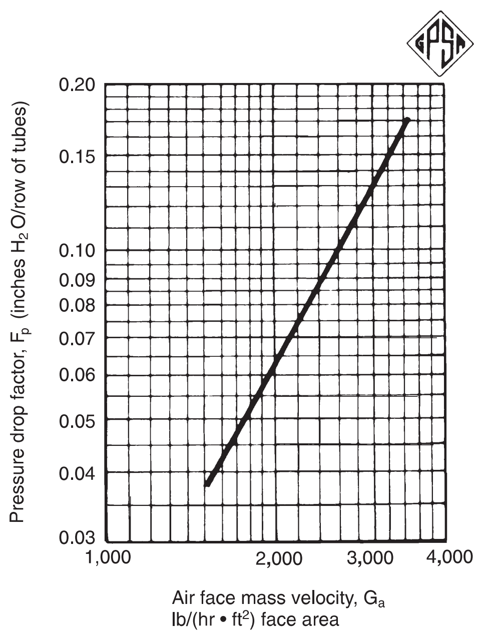


FIG. 10-19

Correction Factor for Fluid Viscosity Within the Tubes

Correction factor * when $\varphi = \left(\frac{\mu}{\mu_w}\right)^{0.14}$ (See Fig. 10-15)

	Correction Factor, ϕ
1. Hydrocarbon vapor; steam; water	1.0
2. Hydrocarbon liquids (18 to 48 API), MEA/DEA solutions	0.96
3. Water/glycol solutions; heat transfer fluids	0.92
4. Lube oils; heavy petroleum fractions (10 to 18 API)	0.85

* When $N_r < 17$, $\varphi = (\frac{\mu}{\mu_w})^{0.25}$ A Reynolds number of less than 17 is only likely for lube oils or heavy petroleum fractions. The minimum recommended value of ϕ to use in Step 10 is 0.80, even though the calculated value may be lower.

Since noise (Sound Pressure Level) is a function of tip speed, slowing fan rotation can reduce noise.

Unfortunately, a fan's pressure capability decreases with the square of the speed. Therefore, the fan's pressure capability must increase to maintain the required airflow.

To increase the pressure capability of a fan, the fan's solidity ratio must be increased, by adding more blades, or using blades with a wider chord, such as a low-noise blade design. Unfortunately, increasing the number of blades can reduce fan efficiency.

Noise requirements are often more restrictive at night. If this is a consideration, slowing the fan as ambient temperature drops using a variable-speed drive can be one solution to reducing noise. As the nighttime ambient temperature drops, required airflow is reduced, therefore fan speed may be slowed to lower noise levels.

Noise-Related Nomenclature

Decibel:

A number representing relative sound intensity expressed with respect to a reference pressure or power level. The usual reference for sound pressure level is of 20 micro newtons per square meter (20 μ/m^2). A decibel is a logarithm (base 10) of a ratio of the power values abbreviated "dB".

Frequency:

Sound vibration rate per second in Hertz (cycles per second).

Low-Noise Fans:

A fan able to operate at low speed due to its high-pressure capability. Fan pressure capability is a function of its solidity ratio. Therefore, a low-noise fan will generally have more or wider blades than would be required if the fan operated at normal tip speeds.

Octave Bands:

Noise is categorized by dividing it into separate frequency bands of octaves or 1/3 octaves. Generally, 63, 125, 250, 500, 1K, 2K, 4K and 8K center frequencies are used to define noise bands in Hertz (cycles/sec).

Sound Power Level:

Acoustical power (energy) can be expressed logarithmically in decibels with respect to a reference power, usually 10^{-12} watts. The relationship is given as: Sound Power Level.

$$PWL = 10 \log \left(\frac{W}{10^{-12}\,watts}\right) \qquad \textbf{Eq 10-3}$$

Sound power level cannot be measured directly but must be calculated from sound pressure levels (SPL) dB. In metric terms, this is known as Lw.

Sound pressure level, known as SPL, or Lp in metric terminology, is the audible noise given in decibels that relates to intensity at a point some distance from the noise source. It can be related to octave bands or as an overall weighted level dB(A).

Weighted sound levels relate the decibel (loudness) to a frequency. Ears can easily pick up high-frequency noises (both intensity and direction) but are relatively insensitive to low-frequency noise. For a stereo system high-frequency speakers must be very carefully located in a room for best results, but low-frequency bass speakers can be placed anywhere, even out of sight.

There are three basic weighting systems: A, B and C. The "A" system, dB(A), most closely relates to our ear, the "B" sys-

FIG. 10-20

Altitude and Atmospheric Pressures[8]

Altitude above Sea Level		Barometer		Atmospheric Pressure	
meters	feet	mm Hg abs.	in Hg abs.	kPa (abs)	psia
0	0	760.0	29.92	101.325	14.696
153	500	746.3	29.38	99.49	14.43
305	1000	733.0	28.86	97.63	14.16
458	1500	719.6	28.33	95.91	13.91
610	2000	706.6	27.82	94.18	13.66
763	2500	693.9	27.32	92.46	13.41
915	3000	681.2	26.82	90.80	13.17
1068	3500	668.8	26.33	89.15	12.93
1220	4000	656.3	25.84	87.49	12.69
1373	4500	644.4	25.37	85.91	12.46
1526	5000	632.5	24.90	84.32	12.23
1831	6000	609.3	23.99	81.22	11.78
2136	7000	586.7	23.10	78.19	11.34
2441	8000	564.6	22.23	75.22	10.91
2746	9000	543.3	21.39	72.39	10.50
3050	10 000	522.7	20.58	69.64	10.10

Reprinted with permission of John M. Campbell and Company.

tem, dB(B), has some specific uses and the "C" system, dB(C), is considered unweighted.

The dB(A) is the most common weighting system. It expresses sound levels with a single number instead of having to specify the noise of each octave band.

Note that the sound range of ACHEs (at close range) is typically between 80 and 105 dB(A).

ACHE Noise

Whether the concern is for overall plant noise or the noise exposure of plant workers in the vicinity of the fans, a different type of noise specification must be used.

Overall, noise limitations from an ACHE are typically a sound power level (PWL) specification for each unit. This limits the contribution of each unit (typically two fans) to the plant noise as a whole. This is usually needed where noise at the plant boundary is considered. Contributions of each part of the plant must be carefully controlled if overall plant noise is limited. PWLs can be expressed as weighted level dB(A) or sometimes even by limitations on each octave band.

If worker protection is the main concern, a limitation of sound pressure level at 3 ft or 1 m below the bundle will probably be imposed as "SPL dB(A) at 3 ft". The OSHA limitation is 90 dB(A) for eight-hour exposure, but 85 dB(A) down to 80 dB(A) is not uncommon.

Predicting Fan Noise

Each fan manufacturer has proprietary equations for predicting fan noise. API Guidelines use the general formula:

$$PWL = 56 + 30 \log \left(\frac{FPM}{1000}\right) + \log HP \qquad \textbf{Eq 10-4}$$

This calculates PWL as dB(A)

Proprietary noise equations are based on actual tests at various speeds and operating conditions considering the following effects:

- Fan diameter
- Fan tip speed
- Blade Type
- Blade pitch angle
- Inlet conditions
- Horsepower

Note: Logs are common logs (base 10).

For example:

14 ft fan

237 RPM (10,424 FPM tip speed)

25.1 HP

Find sound power level

PWL= 56 + 30 log 10.4 + log 25.1

= 100.5 dB(A)

- When considering multiple noise sources (fans) use the relation:

$$PWL_N = PWL + 10 \log N \qquad \textbf{Eq 10-5}$$

The sound power level for 2 adjacent fans is the PWL of one fan plus 10 log 2 or $PWL_2 = PWL + 3$.

A doubling of the noise source adds 3 dB.

- Noise attenuates with distance by the equation:

$$SPL \text{ (at distance R)} = PWL - 20 \log R \qquad \textbf{Eq 10-6}$$

Where R is in feet from the center of the source. Measure R as a "line of sight" distance.

Consider the noise an observer at grade hears at 50 ft from an operating ACHE with the fan on and the line-of-sight distance from grade to the center of the fan is actually 62 ft. Remember what the ear hears is SPL, the noise energy is PWL.

Assume PWL = 100.5 dB(A).

SPL = 100.5 - 20 log 62 = 64.7 dB(A)

This also assumes background noise is at least 10 dB quieter.

Note: If both fans were running, the SPL would have been 67.7 dB(A).

When considering the noise at 3 ft beneath the unit, the drive system and motor noise become dominant at lower tip speeds.

Factors that influence this noise are:

- Motor noise
- Belt or gear noise
- Bearing noise
- Reflected noise from supports
- Background noise

Gear noise is especially significant in a forced draft unit.

Noise Testing

Frequently, the ACHE must be tested for confirmation that its noise does not exceed specifications imposed by the purchaser. There are two basic types of tests normally performed before shipment:

a) Measure SPL dB(A) (Sound Pressure Level) at "3 ft below the bundle or fan guard" — depending on whether the unit is induced or forced draft.

b) Measure PWL (Sound Power Level) using "hemispherical power level test." PWL is specified as either a dB(A) weighted value or by octave bands.

The "SPL at 3 ft" test is by far the most common and least expensive. Usually, only one or two measurements are required. The answer is immediate and read directly from the noise meter.

The hemispherical test is far more complicated and expensive. Several hours, many technicians and a large crane are required to perform this test. Full details of the test are given in API Recommended Practice 631 M, June, 1981.[7]

REFERENCES

1. A.P.I. Standard 661, "Air Cooled Heat Exchangers for General Refinery Services."
2. Briggs, D. E., Young, E. H., "Convection Heat Transfer and Pressure Drop of Air Flowing Across Triangular Pitch of Tubes," Chemical Engineering Progress Symposium Series, Volume 59, No. 41, 1963.
3. Cook, E. M., "Air Cooled Heat Exchangers," Chemical Engineering, May 25, 1964, p. 137; July 6, 1964, p. 131; and August 3, 1964, p. 97.
4. Gardner, K. A., "Efficiency of Extended Surfaces," Trans ASME, Volume 67, 1945, pp. 621-631.
5. Robinson, K. K., Briggs, D. E., "Pressure Drop of Air Flowing Across Triangular Pitch Banks of Finned Tubes," Chemical Engineering Progress Symposium Series, Volume 62, No. 64, 1966.
6. Rubin, Frank L., "Winterizing Air Cooled Heat Exchangers," Hydrocarbon Processing, October 1980, pp. 147-149.
7. API Standard 631 M, First Edition, "Measurement of Noise from Air-Cooled Heat Exchangers," June, 1981.
8. John M. Campbell Co. "Gas Conditioning and Processing, Vol. 2," Eighth Edition.