The Basics of
AIR-COOLED HEAT EXCHANGERS

HUDSON
Products Corporation
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FOREWARD

This brochure is designed to familiarize user with the types, components, and features of Air-Cooled Heat Exchangers (ACHEs) by means of simplified explanations and procedures. It discusses the advantages and disadvantages of each type of help designers become more discriminating, competent, and confident as users of this equipment.

The brochure also provides a method of estimating ACHE size, weight, price and power consumption in the planning stage, but is not intended to provide information sufficient for detailed and final design. If specific assistance is needed, please contact Hudson Products Corporation at (281) 275-8100.

NOMENCLATURE

English Letter Symbols

a	=	heat transfer surface area per unit length of tube	ft^2/ft
A	=	total exchanger bare tube heat transfer surface	ft^2
Aw	=	average wall thickness	in
BWG	=	Birmingham wire gauge	
cp	=	specific heat	Btu/(lb•°F)
Cair	=	$C_{cold} = Q / \Delta t = Q / (t_2\text{-}t_1)$ = air-side heat capacity rate	Btu/(hr•°F)
	=	1.08 • FV • L • W	
Ctube	=	$C_{hot} = Q / \Delta T = Q / (T_1\text{-}T_2)$ = tube-side heat capacity rate	Btu/(hr•°F)
	=	Mc_p	
Cmin	=	minimum heat capacity rate	Btu/(hr•°F)
Cmax	=	maximum heat capacity rate	Btu/(hr•°F)
CMTD	=	corrected mean temperature difference	°F
	=	F • LMTD	
E	=	exchanger thermal effectives	Dimensionless
	=	$\dfrac{C_{hot} \bullet (T_1\text{-}T_2)}{C_{min} \bullet (T_1\text{-}t_1)}$	
	=	$\dfrac{C_{cold} \bullet (t_2\text{-}t_1)}{C_{min} \bullet (T_1\text{-}t_1)}$	
F	=	MTD correction factor	Dimensionless
f	=	Fanning friction factor	Dimensionless

FA	=	face area	ft^2
FV	=	standard air face velocity	std ft/min
G	=	mass velocity	$lb/(sec \bullet ft^2)$
h	=	individual heat transfer coefficient	$Btu/(hr \bullet ft^2 \bullet °F)$
ID	=	inside diameter of tube	ft
K	=	thermal conductivity	$Btu/(hr \bullet ft \bullet °F)$
k	=	parameter, NTU or NTU • R	Dimensionless
	=	$\frac{n \bullet N \bullet a}{1.08 \bullet FV \bullet (l / U)}$	
L	=	tube length	ft
L_E	=	calmed length	ft
LMTD	=	log mean temperature difference	°F
M	=	mass flow rate	lb/hr
mw	=	minimum wall thickness	in
n	=	tubes per row, per foot of exchanger width	l/ft
N	=	rows of tubes in direction of air flow	Dimensionless
N_{Re}	=	Reynolds Number	Dimensionless
Ntu	=	number of heat transfer units	Dimensionless
OD	=	outside diameter of tube	ft
P	=	number of tube-side passes	Dimensionless
Q	=	total exchanger heat load (duty)	Btu/hr
r	=	individual heat transfer resistance	$(hr \bullet ft^2 \bullet °F)/Btu$
R	=	C_{min} / C_{max} = heat capacity rate ratio	Dimensionless
S	=	specific gravity	Dimensionless
t	=	air temperature	°F
T	=	hot fluid temperature	°F
T_{avg}	=	bulk average temperature	°F
U	=	overall heat transfer coefficient (rate)	$Btu/(hr \bullet ft^2 \bullet °F)$
	=	$\frac{1}{(r_i + r_{air} + r_f + r_m)}$	
W	=	width of exchanger	ft
Z	=	parameter, E or E • R	Dimensionless
	=	$\frac{T_1 - T_2}{T_1 - t_1}$	

Greek Letter Symbols

μ	=	viscosity	centipoise
ρ	=	density	lb/ft^3

Subscripts

air	=	air side	max	=	maximum
cold	=	cold fluid = air	min	=	minimum
f	=	tube-side fouling	t	=	total
hot	=	hot fluid = tube-side fluid	1	=	inlet
i	=	tube side	2	=	outlet
m	=	tube metal			

AIR COOLED HEAT EXCHANGERS

A proven means for cooling in the process and power industries

I. DESCRIPTION OF AIR-COOLED HEAT EXCHANGERS

An ACHE is a device for rejecting heat from a fluid directly to ambient air. This is in contrast to rejecting heat to water and then rejecting it to air, as with a shell and tube heat exchanger and a wet cooling tower system.

The obvious advantage of an ACHE is that it does not require water, which means that plants requiring large cooling capacities need not be located near a supply of cooling water.

An ACHE may be as small as an automobile radiator or large enough to reject the heat of turbine exhaust steam condensation from a 1,200 MW power plant – which would require 42 modules, each 90 feet wide by 180 feet long and served by two 60-foot diameter fans driven by 500-horsepower motors.

Components

An ACHE consists of the following components:

(See Figure 1):

- One or more bundles of heat transfer surface.
- An air-moving device, such as a fan, blower, or stack.
- Unless it is natural draft, a driver and power transmission to mechanically rotate the fan or blower.
- A plenum between the bundle or bundles and the air-moving device.
- A support structure high enough to allow air to enter beneath the ACHE at a reasonable rate.
- Optional header and fan maintenance walkways with ladders to grade.
- Optional louvers for process outlet temperature control.
- Optional recirculation ducts and chambers for protection against freezing or solidification of high pour point fluids in cold weather.
- Optional variable pitch fan hub for temperature control and power savings.

Tube Bundle

A tube bundle is an assembly of tubes, headers, side frames, and tube supports as shown in Figure 2. Usually the tube surface exposed to the passage of air has extended surface in the form of fins to compensate for the low heat transfer rate of air at atmospheric pressure and at a low enough velocity for reasonable fan power consumption.

The prime tube is usually round and of any metal suitable for the process, due consideration being given to corrosion, pressure, and temperature limitations. Fins are helical or plate type, and are usually of aluminum for reasons of good thermal conductivity and economy of fabrication. Steel fins are used for very high temperature applications.

Typical Components of an Air-Cooled Heat Exchanger

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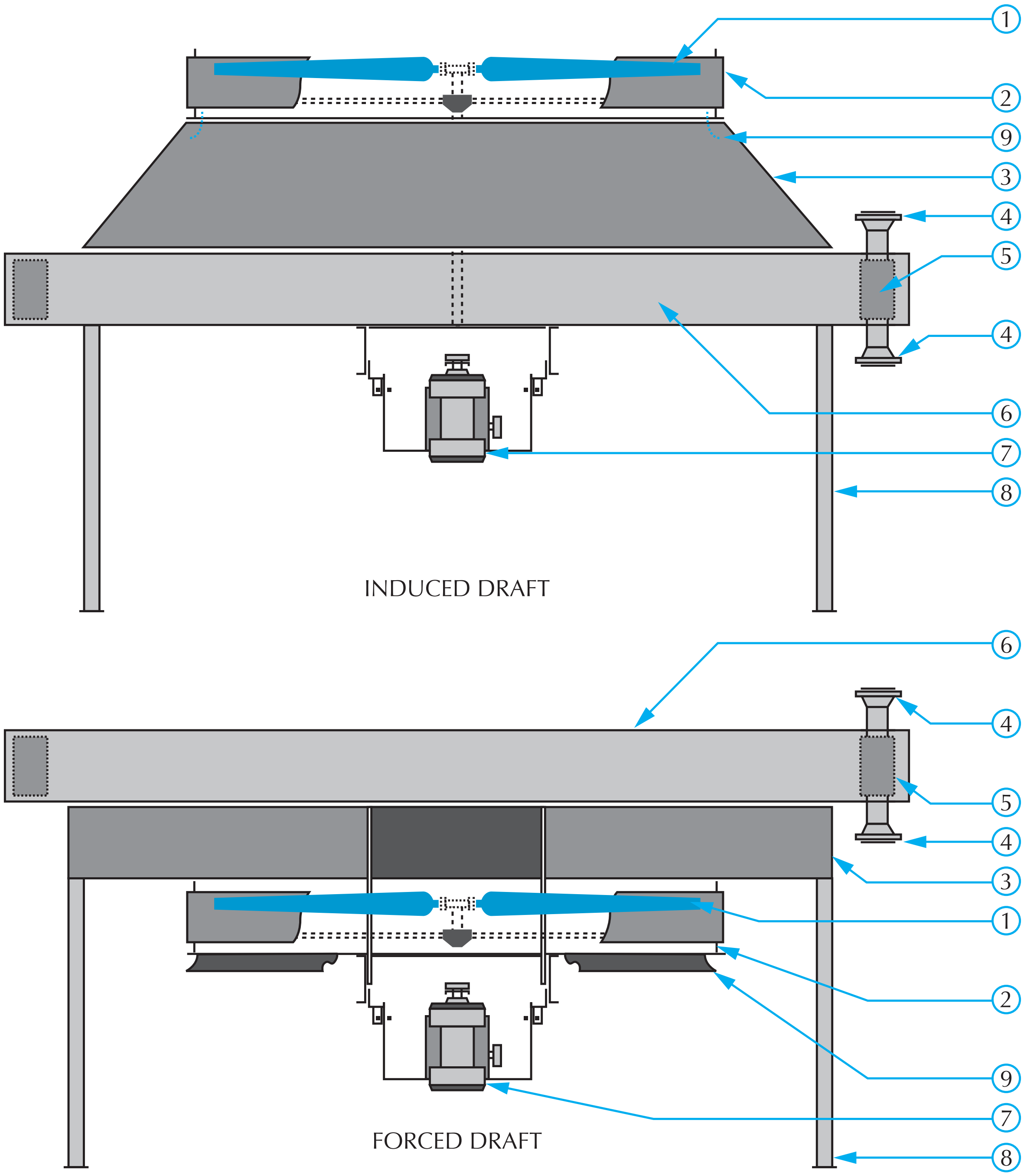
Figure 1



1. Fan
2. Fan Ring
3. Plenum
4. Nozzle
5. Header
6. Tube Bundle
7. Drive Assembly
8. Column Support
9. Inlet Bell

Typical Construction of Tube Bundles with Plug and Cover Plate Headers

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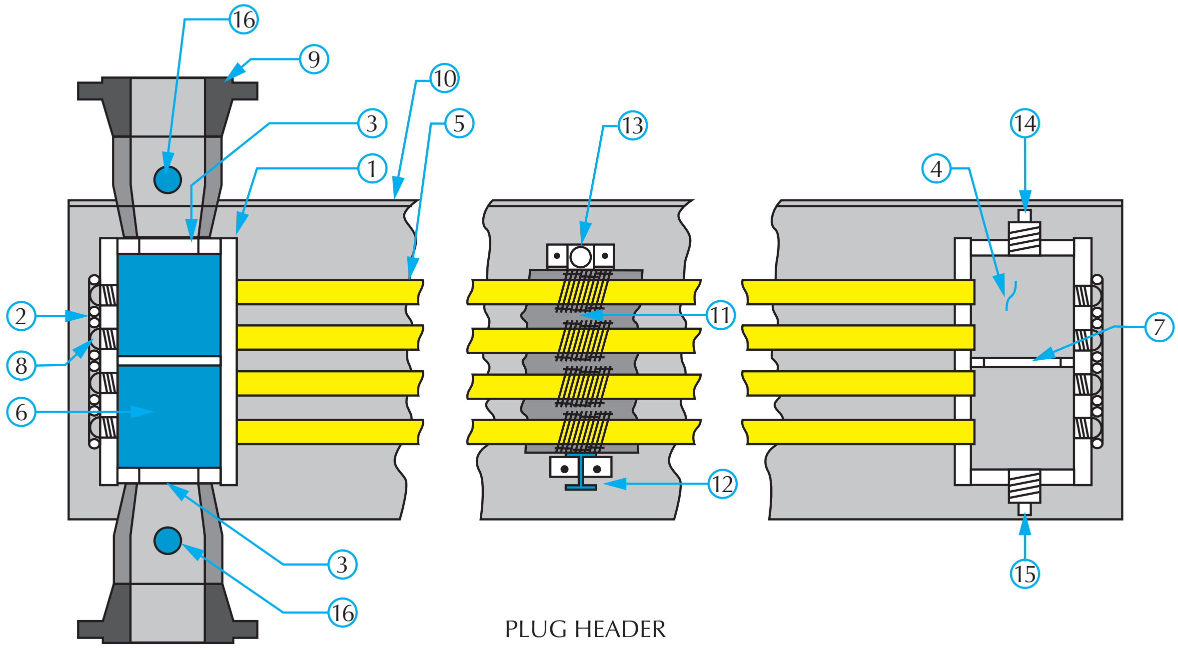
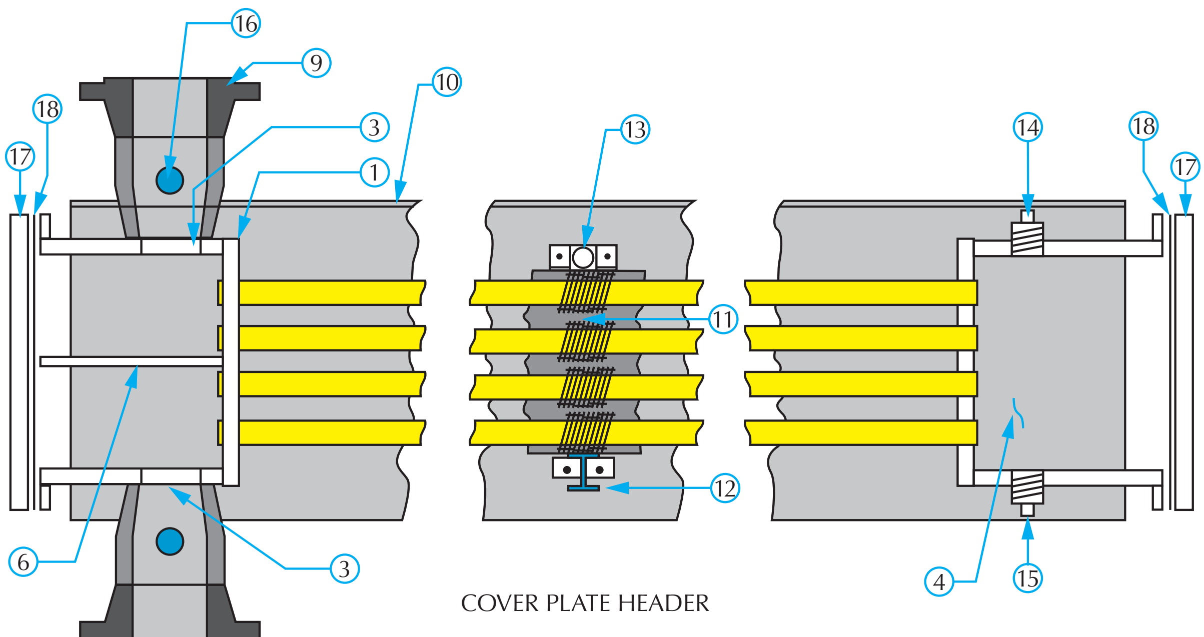
Sugar Land, Texas, USA
Figure 2



PLUG HEADER



COVER PLATE HEADER

1. Tube Sheet
2. Plug Sheet
3. Top and Bottom Plates
4. End Plate
5. Tube
6. Pass Partition
7. Stiffener
8. Plug
9. Nozzle
10. Side Frame
11. Tube Spacer
12. Tube Support Cross-member
13. Tube Keeper
14. Vent
15. Drain
16. Instrument Connection
17. Cover Plate
18. Gasket

Fins are attached to the tubes in a number of ways:

1) by an extrusion process in which the fins are extruded from the wall of an aluminum tube that is integrally bonded to the base tube for the full length.

2) by helically wrapping a strip of aluminum to embed it in a pre-cut helical groove and then peening back the edges of the groove against the base of the fin to tightly secure it, or

3) by wrapping on an aluminum strip that is footed at the base as it is wrapped on the tube. Figure 3 shows a cutaway view of these finned tubes.

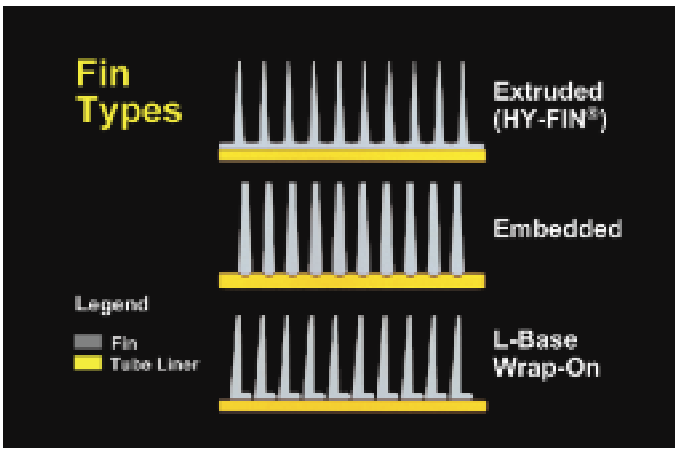


Figure 3

Sometimes serrations are cut in the fins. This causes an interruption of the air boundary layer, which increases turbulence which in turn increases the air-side heat transfer coefficient with a modest increase in the air-side pressure drop and the fan horsepower.

The choice of fin types is critical. This choice is influenced by cost, operating temperatures, and the atmospheric conditions. Each type has different heat transfer and pressure drop characteristics. The extruded finned tube affords the best protection of the liner tube from atmospheric corrosion as well as consistent heat transfer from the initial installation and throughout the life of the cooler. This is the preferred tube for operating temperatures up to 600°F. The embedded fin also affords a continued predictable heat transfer and should be used for all coolers operating above 600°F and below 750°F. The wrap-on footed fin tube can be used below 250°F; however, the bond between the fin and the tube will loosen in time and the heat transfer is not predictable with certainty over the life of the cooler. It is advisable to derate the effectiveness of the wrap-on tube to allow for this probability.

There are many configurations of finned tubes, but manufacturers find it economically practical to limit production to a few standard designs. Tubes are manufactured in lengths from 6 to 60 feet and in diameters ranging from 5/8 inch to 6 inches, the most common being 1 inch. Fins are commonly helical, 7 to 11 fins per inch, 5/16 to 1 inch high, and 0.010 to 0.035 inch thick. The ratio of extended to prime surface varies from 7:1 to 25:1. Bundles are rectangular and typically consist of 2 to 10 rows of finned tubes arranged on triangular pitch. Bundles may be stacked in depths of up to 30 rows to suit unusual services. The tube pitch is usually between 2 and 2.5 tube diameters. Net free area for air flow through bundles is about 50% of face area. Tubes are rolled or welded into the tube sheets of a pair of box headers.

The box header consist of tube sheet, top, bottom, and end plates, and a cover plate that may be welded or bolted on. If the cover is welded on, holes must be drilled and threaded opposite each tube for maintenance of the tubes. A plug is screwed into each hole, and the cover is called the plug sheet. Bolted removable cover plates are used for improved access to headers in severe fouling services. Partitions are welded in the headers to establish the tube-side flow pattern, which generates suitable velocities in as near countercurrent flow as possible for maximum mean temperature difference. Partitions and stiffeners (partitions with flow openings) also act as structural stays. Horizontally split headers may be required to accommodate differential tube expansion in services having high fluid temperature differences per pass. Figure 4 illustrates common heat types.

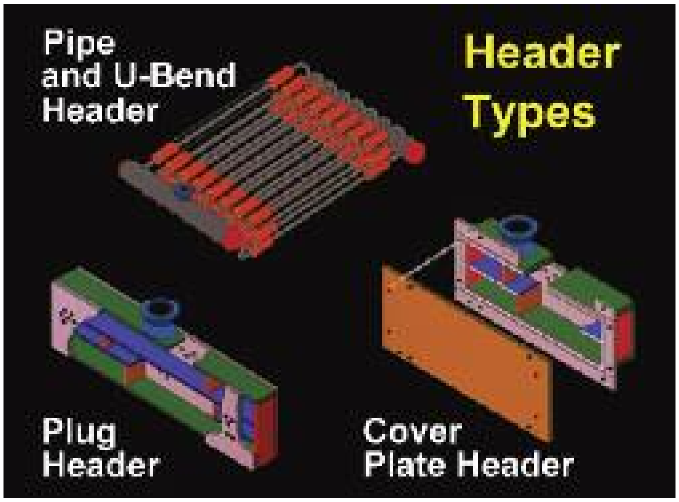


Figure 4

Figure 5

Bundles are usually arranged horizontally with the air entering below and discharging vertically. Occasionally bundles are arranged vertically with the air passing across horizontally, such as in a natural draft tower where the bundles are arranged vertically at the periphery of the tower base. Bundles can also be arranged in an "A" or "V" configuration, the principle advantage of this being a saving of plot area. The disadvantages are higher horsepower requirements for a given capacity and decreased performance when winds on exposed sides inhibit air flow.

With practical limits, the longer the tubes and the greater the number of rows, the less the heat transfer surface costs per square foot. One or more bundles of the same or differing service may be combined in one unit (bay) with one set of fans. All bundles combined in a single unit will have the same air-side static pressure loss. Consequently, combined bundles having different numbers of rows must be designed for different velocities.

Axial Flow Fans

Figure 5 displays the air moving device for an ACHE which is commonly an axial flow, propeller type fan that either forces the air across the bundles (forced draft) or pulls it across (induced draft). To provide redundancy in case a mechanical unit fails and to provide the basic control achievable by running one fan or two, a bundle or set of bundles is usually provided with two fans.

Even distribution of the air across the tube bundle is critical for predictable, uniform heat transfer. This is achieved by adequate fan coverage and static pressure loss across the bundle. Good practice is to keep the fan projected area to a minimum of 40% of the projected face area of the tube bundle and the bundle static pressure loss at least 3.5 times the velocity pressure loss through the fan ring. For a two fan unit this is generally assured if the ratio of tube length to bundle width is in the range of 3 to 3.5 and the number of tube rows is held to 4 rows minimum with the net free area for air flow at about 50% of the face area of the bundle.

Fans can very in size from 3 to 60 feet in diameter and can have from 2 to 20 blades. Blades can be made of wood, steel, aluminum, or fiberglass-reinforced plastic, and can be solid or hollow. Blades can have straight sides or be contoured. The more efficient type has a wide chord near the center and tapers to a narrow chord at the tip, with a slight twist. The twist and taper compensate for the slower velocity of the blade nearer the center to produce a uniform, efficient air velocity profile.

Fans may have fixed or adjustable pitch blades. Except for small diameters (less than 5 feet), most ACHEs have adjustable pitch blades. Adjustable pitch fans are manufactured in two types. One is manually adjustable (with the fans off) and the other is automatically adjustable (while running). Most automatically adjustable pitch fans change their pitch by means of pneumatically actuated diaphragm working against large springs inside the hub.

Plenum

The air plenum is an enclosure that provides for the smooth flow of air between the fan and bundle. Plenums can be box type or slopesided type. The slopesided type gives the best distribution of air over the bundles, but is almost exclusively used with induced draft because hanging a machinery mount from a slopesided forced draft plenum presents structural difficulties.

Mechanical Equipment

Fans may be driven by electric motors, steam turbines, gas or gasoline engines, or hydraulic motors. The overwhelming choice is the electric motor. Hydraulic motors are sometimes used when power from an electric utility is unavailable. Hydraulic motors also provide variable speed control, but have low efficiencies.

The most popular speed reducer is the high-torque positive type belt drive, which uses sprockets that mesh with the timing belt cogs. They are used with motors up to 50 or 60 horsepower, and with fans up to about 18 feet in diameter. Banded V-belts are still often used in small to medium sized fans, and gear drives are used with very large motors and fan diameters. Fan speed is set by using a proper combination of sprocket or sheave sizes with timing belts or V-belts, and by selecting a proper reduction ratio with gears. Fan tip speed should not be above 12,000 feet per minute for mechanical reasons, and may be reduced to obtain lower noise levels. Motor and fan speed is sometimes controlled with variable frequency drives. Figure 6 provides a breakdown of the mechanical equipment.

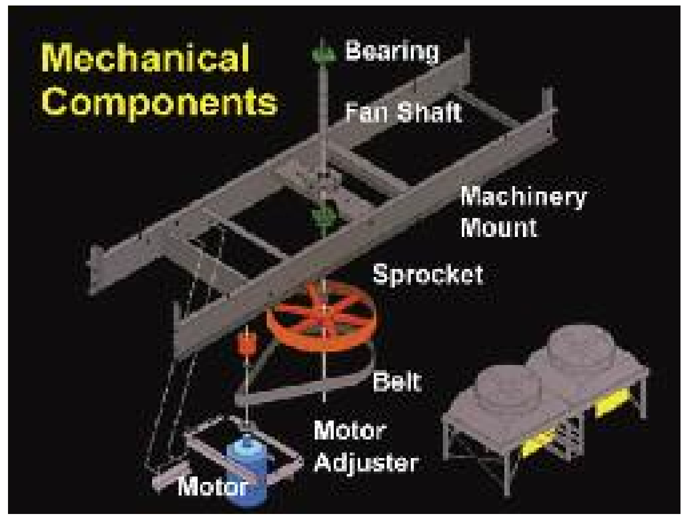


Figure 6

Structure

The structure consists of columns, braces, and cross beams that support the exchanger at a sufficient elevation above grade to allow the necessary volume of air to enter below at an approach velocity low enough to allow unimpeded fan performance and to prevent unwanted recirculation of hot air. To conserve ground space in oil refineries and chemical plants, ACHEs are usually mounted above, and supported by, pipe racks with other equipment occupying the space underneath the pipe rack. ACHE structures are designed for appropriate wind, snow, seismic, piping, dead and live loads.

Comparison of Induced and Forced Draft Units

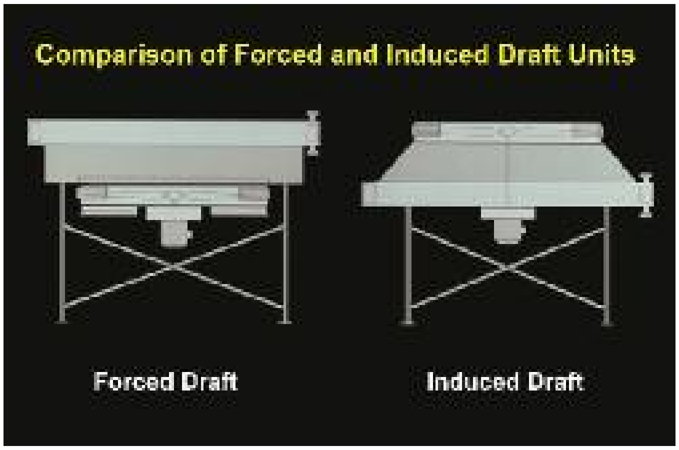


Figure 7

Induced Draft

Advantages:

1) Better distribution of air across the bundle.

2) Less possibility of hot effluent air recirculating into the intake. The hot air is discharged upward at approximately 2.5 times the intake velocity, or about 1,500 feet per minute.

3) Better process control and stability because the plenum covers 60% of the bundle face area, reducing the effects of sun, rain, and hail.

4) Increased capacity in the fan-off or fan failure condition, since the natural draft stack effect is much greater.

Disadvantages and Limitations:

1) Possibly higher horsepower requirements if the effluent air is very hot.

2) Effluent air temperature should be limited to 220°F to prevent damage to fan blades, bearings, or other mechanical equipment in the hot air stream. When the process inlet temperature exceeds 350°F, forced draft design should be considered because high effluent air temperatures may occur during fan-off or low air flow operation.

3) Fans are less accessible for maintenance, and maintenance may have to be done in the hot air generated by natural convection.

4) Plenums must be removed to replace bundles.

Forced Draft

Advantages:

1) Possibly lower horsepower requirements if the effluent air is very hot. (Horsepower varies inversely with the absolute temperature.)

2) Better accessibility of fans and upper bearings for maintenance.

3) Better accessibility of bundles for replacement.

Accommodates higher process inlet temperatures.

Disadvantages:

Less uniform distribution of air over the bundle.

Increased possibility of hot air recirculation, resulting from low discharge velocity from the bundles, high intake velocity to the fan ring, and no stack.

Low natural draft capability of fan failure.

Complete exposure of the finned tubes to sun, rain, and hail, which results in poor process control and stability.

In most cases, the advantages of induced draft design outweigh the disadvantages.

II. THERMAL DESIGN

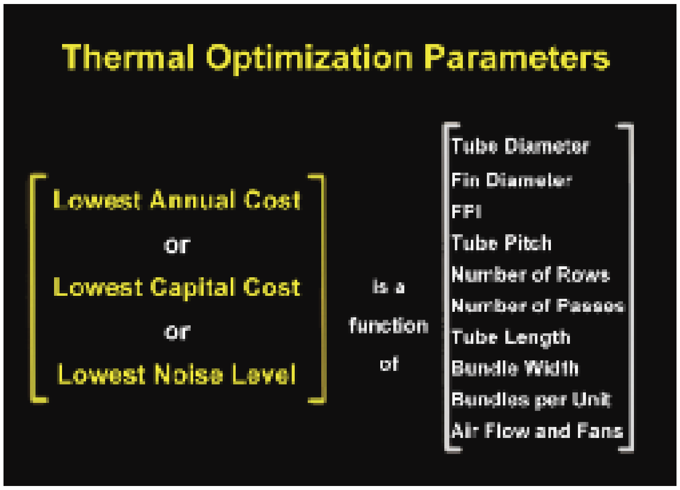


Figure 8

There are more parameters to be considered in the thermal design of ACHEs than for shell and tube exchangers (see Figure 8). ACHEs are subject to wide variety of constantly changing climatic conditions which pose problems of control not encountered with shell and tube exchangers. Designers must achieve an economic balance between the cost of electrical power for the fans and the initial capital expenditure for the equipment. A decision must be made as to what ambient air temperature should be used for design. Air flow rate and exhaust temperature is initially unknown and can be varied in the design stage by varying the number of tube rows and thus varying the face area.

Because the number of tube rows, the face area, the air face velocity, and the geometry of the surface can all be varied, it is possible to generate many solutions to a given thermal problem. However, there is obviously an optimum solution in terms of capital and operating costs.

The basic heat transfer relationships that apply to shell and tube exchangers also apply to ACHEs. The fundamental relation is the Fourier equation:

$$Q = U \bullet A \bullet (T - t)_{mean}$$

Where

$$(T - t)_{mean} = CMTD = LMTD \bullet F$$

$$\frac{(T_1 - t_2) - (T_2 - t_1)}{\ln\left[\dfrac{(T_1 - t_2)}{(T_2 - t_1)}\right]} \bullet F$$

F is a factor that corrects the log mean temperature difference for any deviation from true counter-current flow. In ACHEs, the air flows substantially unmixed upward across the bundles and the process fluid can flow back and forth and downward as directed by the pass arrangement. With four or more downward passes, the flow is considered counter-current; and, so the factor “F” is 1.0. The correction factors for one, two, and three passes, given by Figures 9 – 11, were calculated from the effectiveness values developed by Stevens, Fernandez, and Wolf[1] for the appropriate counter-cross flow arrays.

As is apparent, initially neither the area nor the overall heat transfer rate nor the effluent air temperatures are known. The traditional approach in the design of ACHEs entailed an iterative trial and error procedure both on the CMTD and the transfer rate until the area satisfied both. Specifically, an air rise was assumed, the CMTD was calculated, an overall heat transfer coefficient was assumed, and an exchanger size was selected with the expected necessary area. An appropriate face velocity was then used to calculate an effluent air temperature, and the process was repeated until the assumed effluent air temperature matched the calculated value. The individual coefficients and the overall coefficient were then calculated, and the whole process was repeated until the calculated “U” and CMTD were sufficiently close to the assumed values.

However, there is another method that eliminates trial and error on the CMTD and leaves only the trial and error on the tube-side film coefficient. The following discussion presents the Ntu Method described by Kays and London in *Compact Heat Exchangers*[2], as applied to ACHES.

The following are definitions based on *Compact Heat Exchangers*:

1. Hot fluid heat capacity rate = $C_h = C_{tube}$

$$= (Mc_p)_{tube} = \frac{Q}{T_1 - T_2}$$

2. Cold fluid heat capacity rate = $C_c = C_{air}$

$$= (Mc_p)_{air} = \frac{Q}{t_2 - t_1}$$

3. Number of heat transfer units = Ntu = $\dfrac{A \bullet U}{C_{min}}$

4. Heat capacity rate ratio = R = $\dfrac{C_{min}}{C_{max}}$

5. ACHE heat transfer effectiveness = E

$$E = \frac{C_h (T_1 - T_2)}{C_{min} (T_1 - t_1)} = \frac{C_c (t_2 - t_1)}{C_{min} (T_1 - t_1)}$$

See pages i – iii for additional definitions and nomenclature.

We define air flow in terms of standard cubic feet per minute (scfm) as the product of the effective width and length of the exchanger in feet, and the face velocity (FV) in standard feet per minute (sfm). For any ACHE service, it will not necessarily be apparent at the design stage whether the air or the hot tube-side fluid will have the minimum heat capacity rate, since the mass flow rate of the air is unknown. The two cases presented below will cover both design situations.

MTD Correction Factors / 1 Pass-Cross Flow

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Figure 9

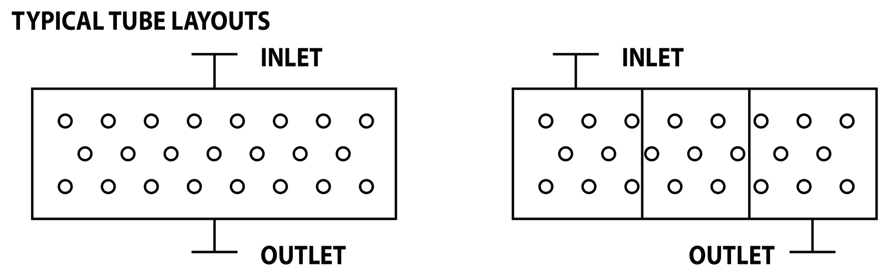
NOMENCLATURE:

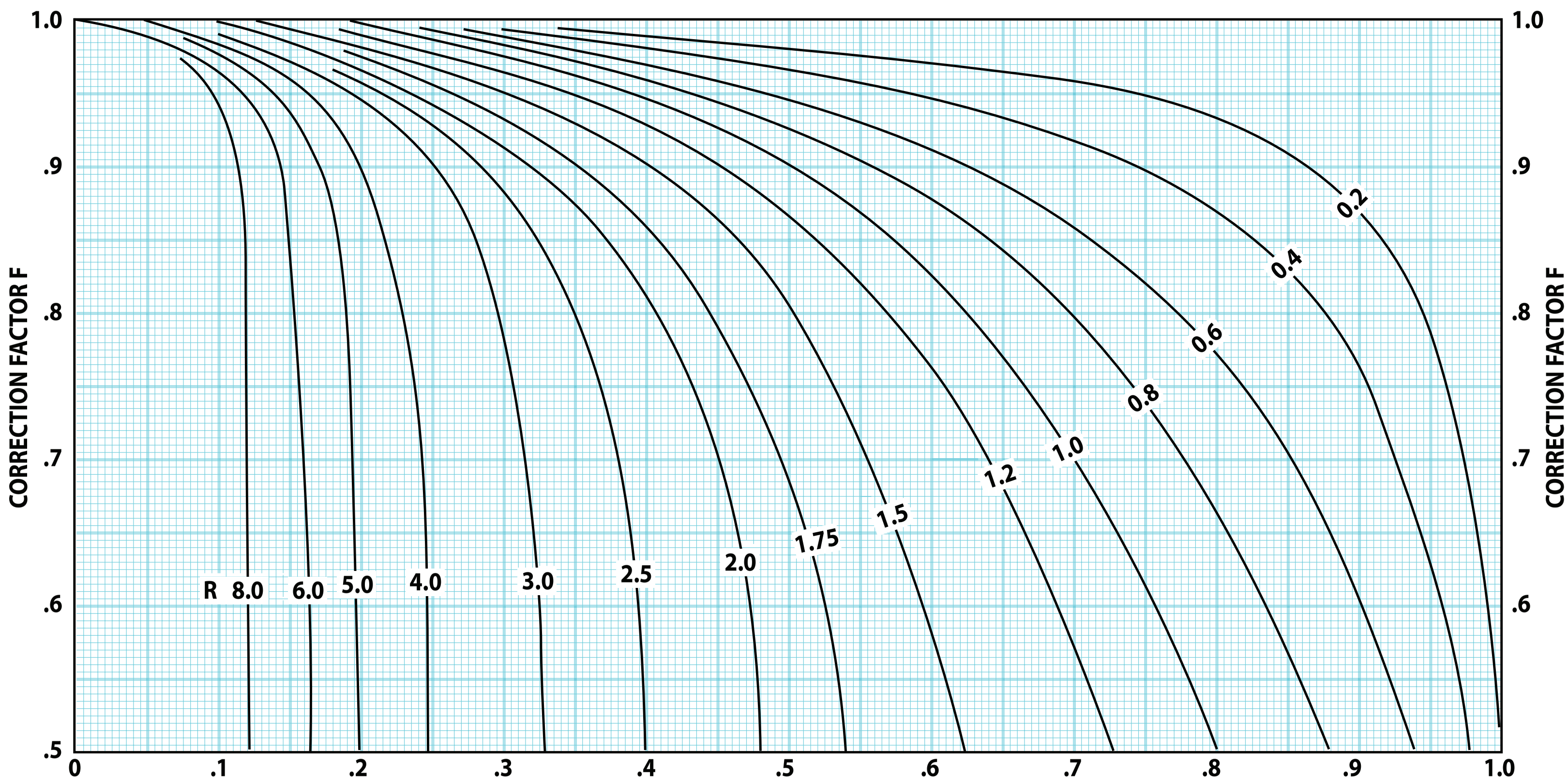
T_1 = INLET TEMPERATURE TUBE SIDE

T_2 = OUTLET TEMPERATURE TUBE SIDE

t_1 = INLET TEMPERATURE AIR SIDE

t_2 = OUTLET TEMPERATURE AIR SIDE





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

MTD CORRECTION FACTORS
1 PASS-CROSS FLOW
BOTH FLUIDS UNMIXED

MTD Correction Factors / 2 Pass-Cross Flow

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Figure 10

NOMENCLATURE:

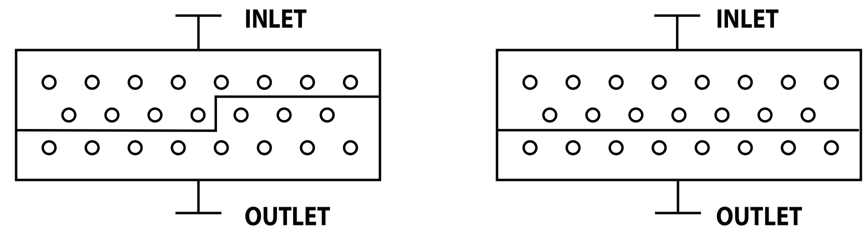
T_1 = INLET TEMPERATURE TUBE SIDE

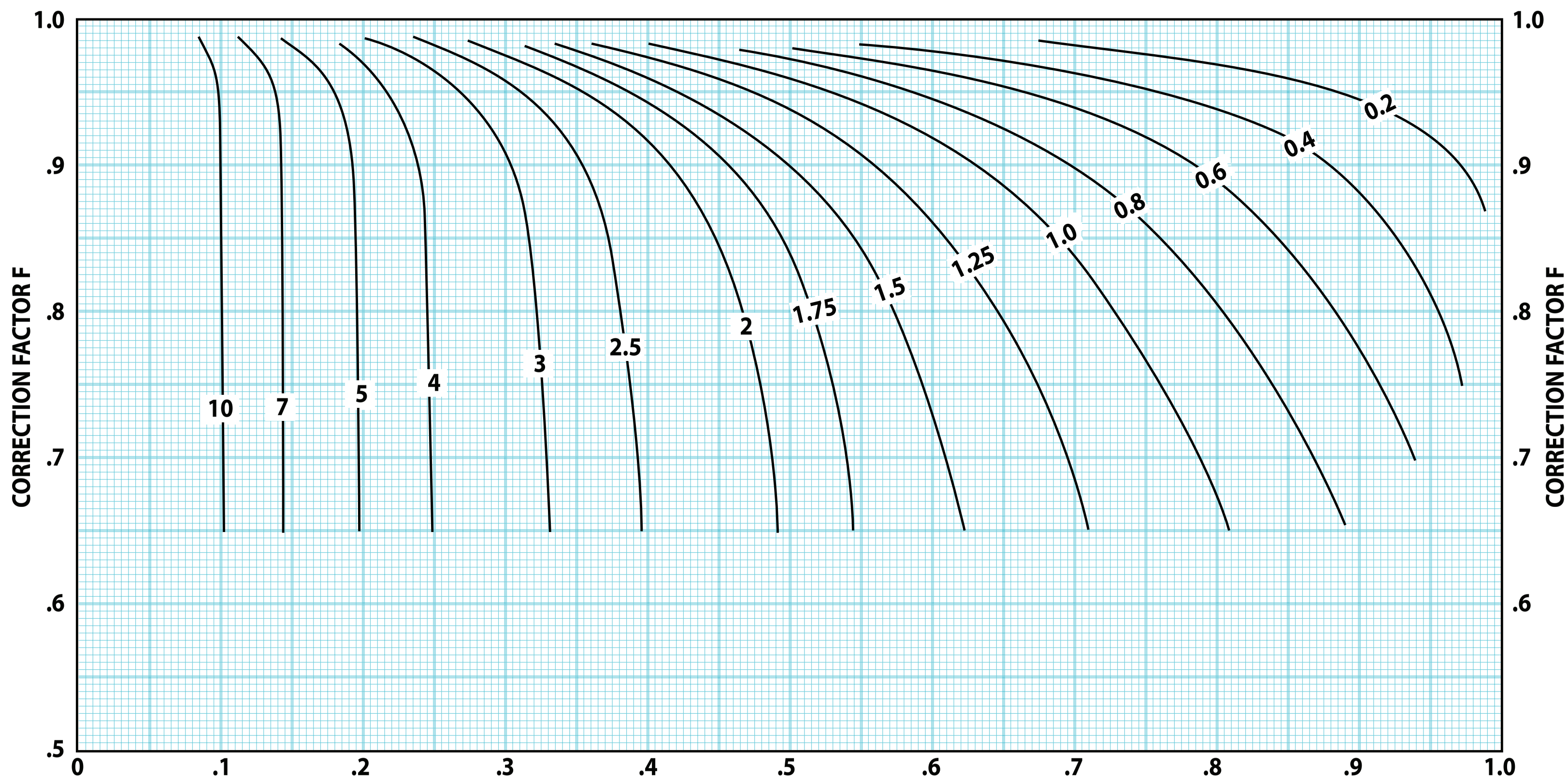
T_2 = OUTLET TEMPERATURE TUBE SIDE

t_1 = INLET TEMPERATURE AIR SIDE

t_2 = OUTLET TEMPERATURE AIR SIDE

TYPICAL TUBE LAYOUTS





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

MTD CORRECTION FACTORS
2 PASS-COUNTER CROSS FLOW
BOTH FLUIDS UNMIXED

MTD Correction Factors / 3 Pass-Cross Flow

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Figure 11

NOMENCLATURE:

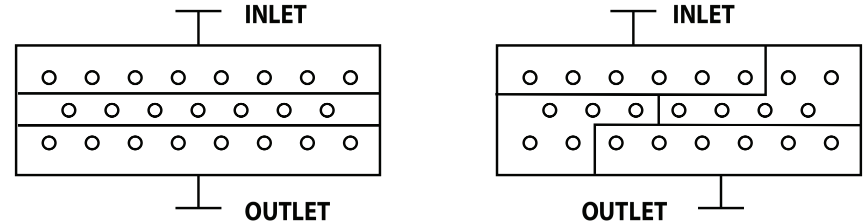
T_1 = INLET TEMPERATURE TUBE SIDE

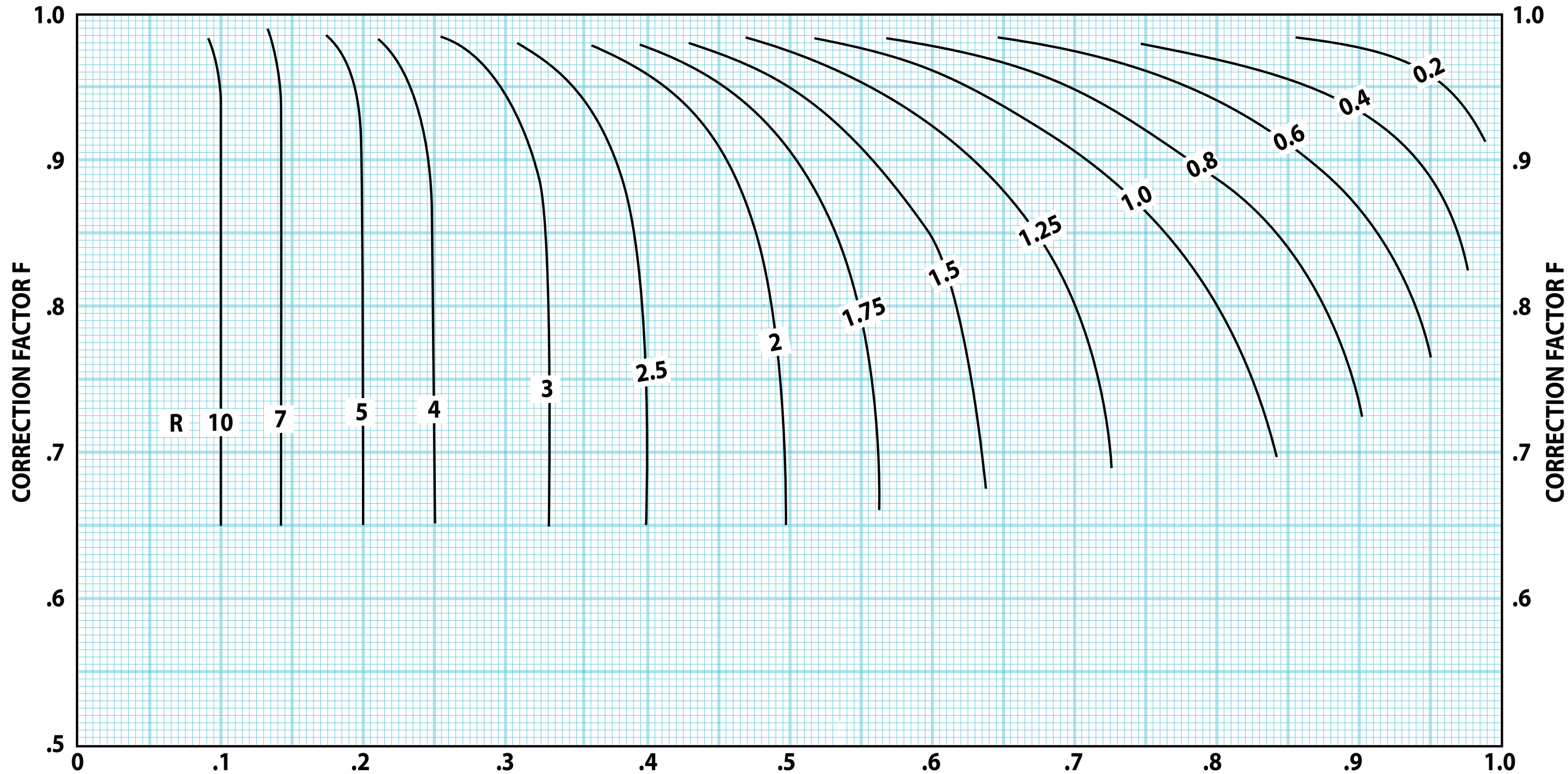
T_2 = OUTLET TEMPERATURE TUBE SIDE

t_1 = INLET TEMPERATURE AIR SIDE

t_2 = OUTLET TEMPERATURE AIR SIDE

TYPICAL TUBE LAYOUTS





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

MTD CORRECTION FACTORS
3 PASS-COUNTER CROSS FLOW
BOTH FLUIDS UNMIXED

CASE I. $C_{min} = C_{air} = C_{cold}$

1. From Definition 4 above:

$$R = \frac{C_{min}}{C_{max}} = \frac{C_{air}}{C_{hot}} = \frac{scfm \bullet 1.08}{\left[\dfrac{Q}{(T_1 - T_2)}\right]}$$

$$= \frac{FV \bullet L \bullet W \bullet 1.08}{\left[\dfrac{Q}{(T_1 - T_2)}\right]}$$

$$= \frac{FV \bullet L \bullet W \bullet 1.08 \bullet (T_1 - T_2)}{Q}$$

Note: 1.08=0.075 lb/ft³ • 60 min/hr · 0.24 Btu/(lb • °F)

2. From Definition 5 above and by substitution for $\dfrac{C_{hot}}{C_{air}}$:

$$E = \frac{Ch\,(T_1 - T_2)}{C_{min}\,(T_1 - t_1)}$$

$$= \frac{Q}{FV \bullet L \bullet W \bullet 1.08 \bullet (T_1 - t_1)}$$

Multiplying Equation 1 by Equation 2:

$$3.\ ER = \frac{FV \bullet L \bullet W \bullet 1.08 \bullet (T_1 - T_2)}{Q}$$

$$\bullet \frac{Q}{FV \bullet L \bullet W \bullet 1.08 \bullet (T_1 - t_1)}$$

$$= \frac{T_1 - T_2}{T_1 - t_1}$$

If we let $\dfrac{T_1 - T_2}{T_1 - t_1} = Z$, then

$Z = E \bullet R$ for Case I

From Definition 3 above:

$$Ntu = \frac{A \bullet U}{C_{min}} = \frac{A \bullet U}{C_{air}} = \frac{n \bullet N \bullet a \bullet W \bullet L \bullet U}{1.08 \bullet W \bullet L \bullet FV}$$

$$= \frac{n \bullet N \bullet a}{1.08 \bullet FV \bullet (r_i + r_{air} + r_f + r_m)}$$

$$\text{Let } k = \frac{n \bullet N \bullet a}{1.08 \bullet FV \bullet (r_i + r_{air} + r_f + r_m)}$$

$k = Ntu$ for Case 1

We can plot the expression for Case I with E • R and Ntu as coordinates and R as the parameter. Knowing that Z = E • R and that k = Ntu, we can find R on the plot.

From Equation 1, Case I:

$$R = \frac{FV \bullet L \bullet W \bullet 1.08 \bullet (T_1 - T_2)}{Q}$$

$$W = \frac{Q \bullet R}{1.08 \bullet FV \bullet L \bullet (T_1 - T_2)}$$

$$t_2 = \frac{(T_1 - T_2)}{R} + t_1$$

CASE II. $C_{min} = C_{tube} = C_{hot}$

From Definition 5 above:

$$E = \frac{Ch\,(T_1 - T_2)}{C_{min}\,(T_1 - t_1)} = \frac{T_1 - T_2}{T_1 - t_1} = Z$$

From Definition 4 above:

$$R = \frac{C_{min}}{C_{max}} = \frac{C_{hot}}{C_{air}}$$

$$= \frac{Q}{FV \bullet L \bullet W \bullet 1.08 \bullet (T_1 - T_2)}$$

$$Ntu = \frac{n \bullet N \bullet a \bullet W \bullet L}{\left[\dfrac{Q}{(T_1 - T_2)}\right] \bullet (r_i + r_{air} + r_f + r_m)}$$

$$R \bullet Ntu = \frac{n \bullet N \bullet a}{1.08 \bullet FV \bullet (r_i + r_{air} + r_f + r_m)}$$

$k = R \bullet Ntu$ for Case II

CASE I or II. Selection Criteria

We can plot the expression with E and $R \bullet Ntu$ as coordinates and R as the parameter on the same graph as Case I, with $R = 1$ common to both plots.

For values of R above the line $R = 1$:

$$W = \frac{Q}{1.08 \bullet R \bullet (T_1 - T_2) \bullet FV \bullet L}$$

and $t_2 = R \bullet (T_1 - T_2) + t_1$

For values of R below the line $R = 1$:

$$W = \frac{Q \bullet R}{1.08 \bullet (T_1 - T_2)\, FV \bullet L}$$

and $t_2 = \dfrac{T_1 - T_2}{R} + t_1$

It can be shown that E, Ntu, and R can be related for any flow arrangement. For counter-current flow, the expression is:

$$E = \frac{1 - e^{-Ntu \bullet (1-R)}}{1 - R \bullet e^{-Ntu \bullet (1-R)}}$$

Figures 12-15 show plots relating these variables for the following cases:

1. Cross flow with both fluids unmixed (one-pass ACHE, Figure 12).
2. Two-pass counter cross flow with both fluids unmixed in each pass but with the hot fluid mixed between passes (two-pass ACHE, Figure 13).
3. Three-pass counter cross flow with both fluids unmixed in each pass but with the hot fluid mixed between passes (three-pass ACHE, figure 14).
4. Counter flow (ACHE with four or more passes, Figure 15).

The case of the isothermal exchanger is much simpler. As R goes to zero, the equation for effectiveness reduces to:

$E = 1 - e^{-Ntu}$

And Ntu is as in Case I for $C_{min} = C_{air}$:

$$Ntu = \frac{n \bullet N \bullet a}{1.08 \bullet FV \bullet (r_i + r_{air} + r_f + r_m)}$$

$$E = \frac{Q}{Q_{max}} = \frac{Q}{1.08 \bullet (T_1 - t_1) \bullet FV \bullet L \bullet E}$$

$$W = \frac{Q}{L \bullet W \bullet FV \bullet 1.08 \bullet (T_1 - t_1)}$$

Air Cooler Sizing Chart / 1 Pass

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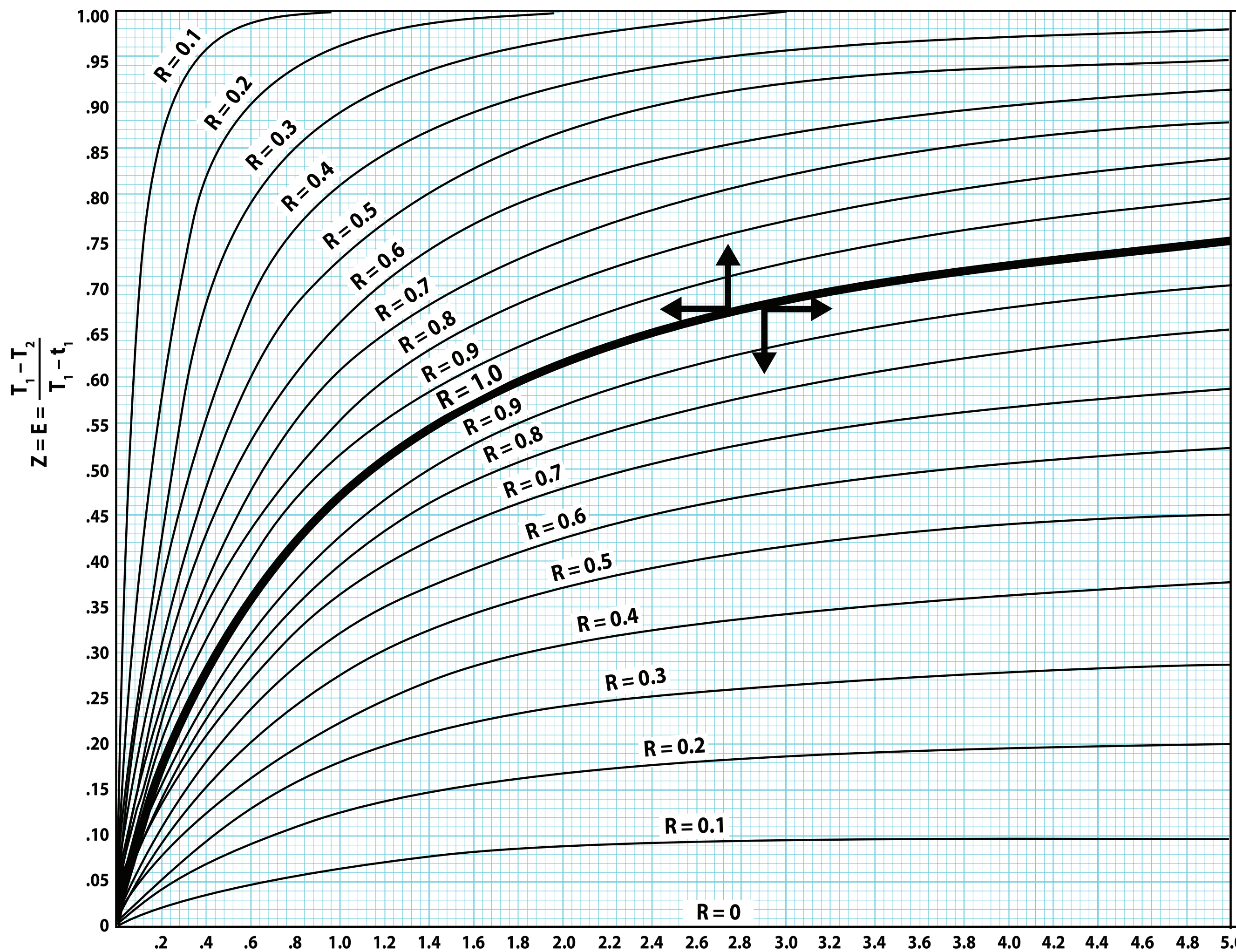
Figure 12

$$k = Ntu = \frac{n \cdot N \cdot a \cdot U}{1.08 \cdot FV} = \frac{n \cdot N \cdot a}{1.08 \cdot FV\,(r_{air} + r_i + r_f + r_m)}$$



$$k = Ntu = \frac{n \cdot N \cdot a \cdot U}{1.08 \cdot FV} = \frac{n \cdot N \cdot a}{1.08 \cdot FV\,(r_{air} + r_i + r_f + r_m)}$$

ONE (1) PASS

FOR R ABOVE R = 1.0

$$FA = \frac{M \cdot cp}{1.08 \cdot FV \cdot R}$$

$$= \frac{Q}{(T_1 - T_2)1.08 \cdot FV \cdot R}$$

$$t_2 = R\,(T_1 - T_2) + t_1$$

$$Z = E \cdot R = \frac{T_1 - T_2}{T_1 - t_1}$$

FOR R BELOW R = 1.0

$$FA = \frac{R \cdot M \cdot cp}{1.08 \cdot FV}$$

$$= \frac{R \cdot Q}{(T_1 - T_2)1.08 \cdot FV}$$

$$t_2 = \frac{(T_1 - T_2)}{R} + t_1$$

Air Cooler Sizing Chart / 2 Pass

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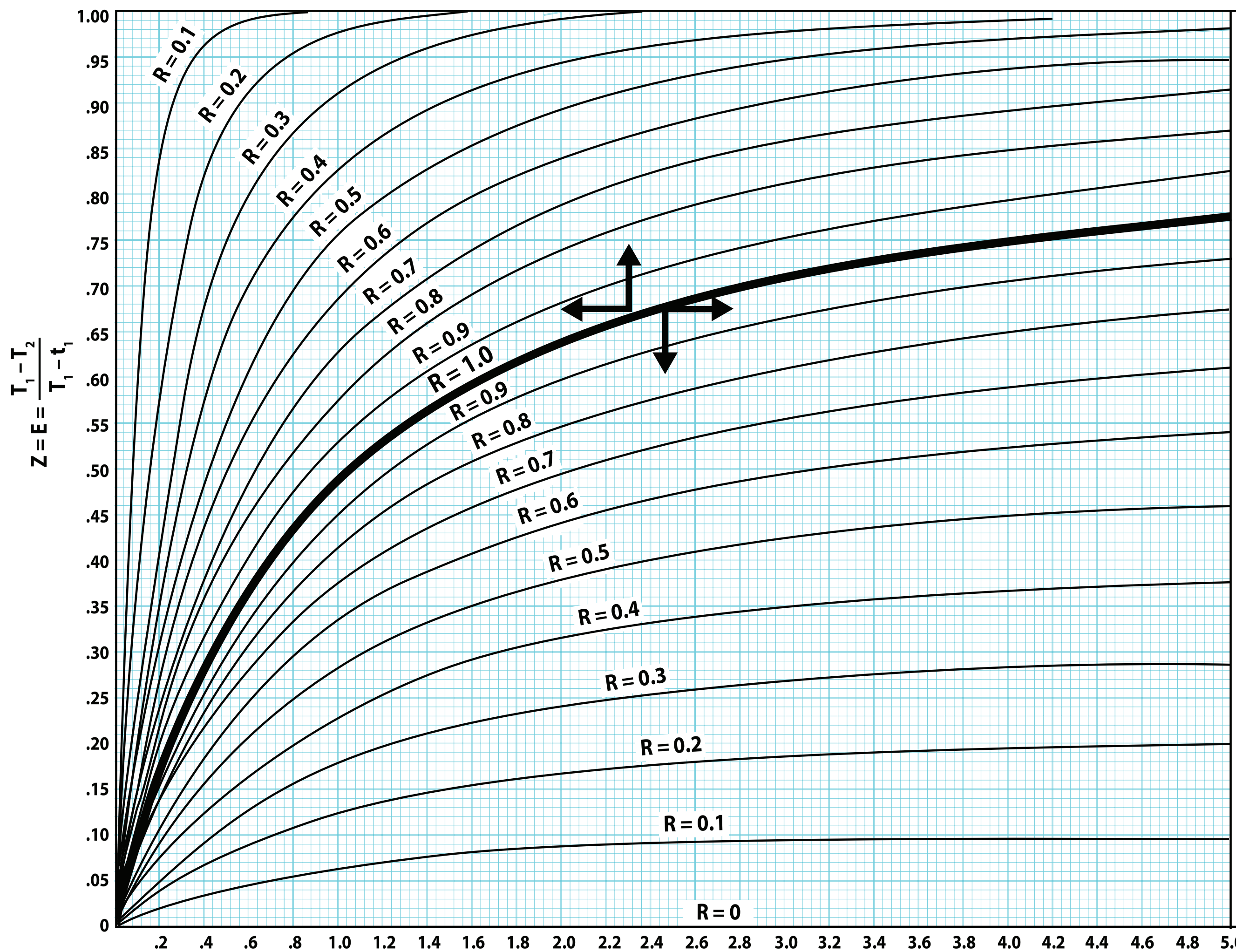
Figure 13

$$k = Ntu = \frac{n \cdot N \cdot a \cdot U}{1.08 \cdot FV} = \frac{n \cdot N \cdot a}{1.08 \cdot FV\,(r_{air} + r_i + r_f + r_m)}$$



$$k = Ntu = \frac{n \cdot N \cdot a \cdot U}{1.08 \cdot FV} = \frac{n \cdot N \cdot a}{1.08 \cdot FV\,(r_{air} + r_i + r_f + r_m)}$$

TWO (2) PASSES

FOR R ABOVE R = 1.0

$$FA = \frac{M \cdot cp}{1.08 \cdot FV \cdot R} = \frac{Q}{(T_1 - T_2)1.08 \cdot FV \cdot R}$$

$$t_2 = R\,(T_1 - T_2) + t_1$$

$$Z = E \cdot R = \frac{T_1 - T_2}{T_1 - t_1}$$

FOR R BELOW R = 1.0

$$FA = \frac{R \cdot M \cdot cp}{1.08 \cdot FV} = \frac{R \cdot Q}{(T_1 - T_2)1.08 \cdot FV}$$

$$t_2 = \frac{(T_1 - T_2)}{R} + t_1$$

Air Cooler Sizing Chart / 3 Pass

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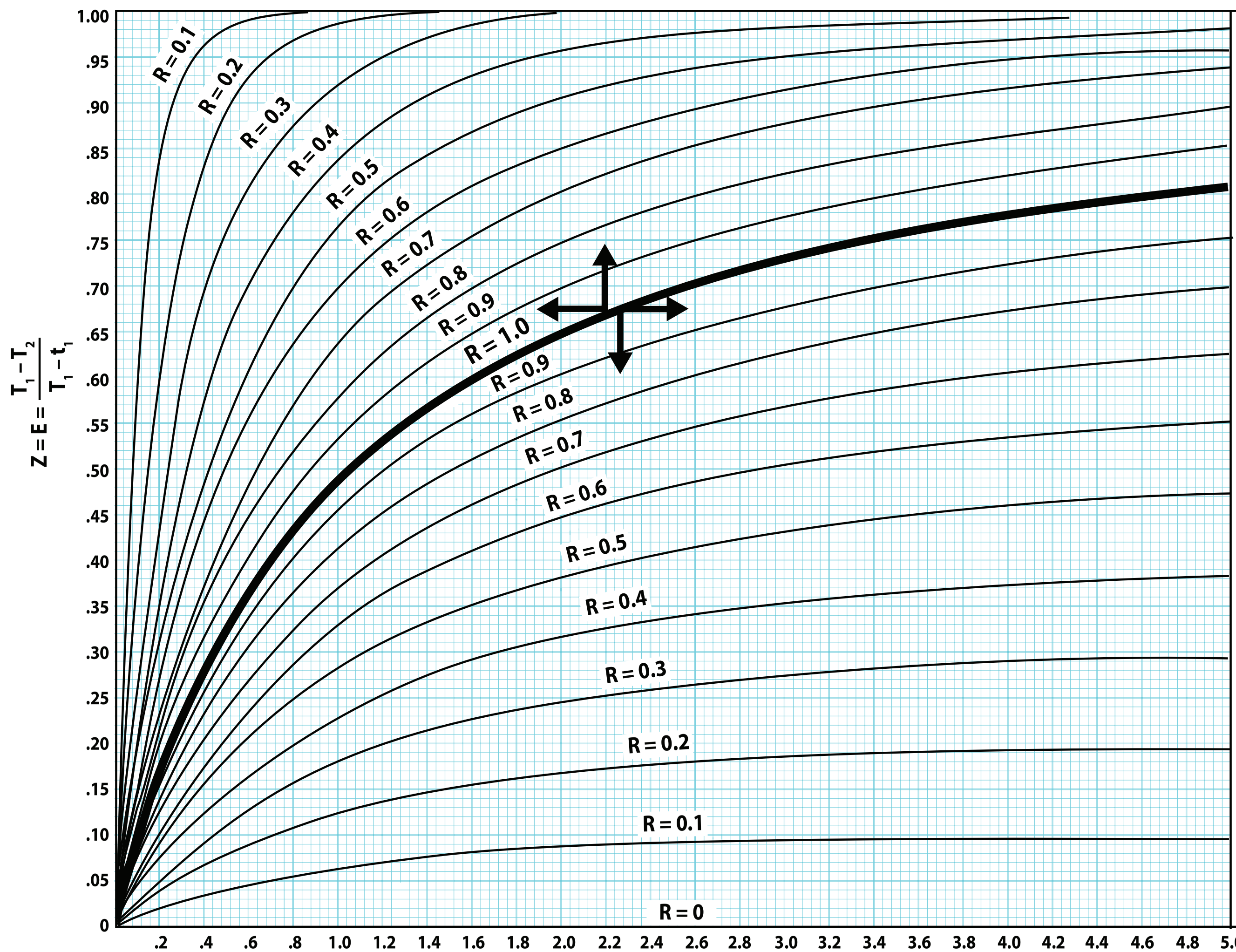
Figure 14

$$k = Ntu = \frac{n \cdot N \cdot a \cdot U}{1.08 \cdot FV} = \frac{n \cdot N \cdot a}{1.08 \cdot FV\,(r_{air} + r_i + r_f + r_m)}$$



$$Z = E = \frac{T_1 - T_2}{T_1 - t_1}$$

$$k = Ntu = \frac{n \cdot N \cdot a \cdot U}{1.08 \cdot FV} = \frac{n \cdot N \cdot a}{1.08 \cdot FV\,(r_{air} + r_i + r_f + r_m)}$$

THREE (3) PASSES

FOR R ABOVE R = 1.0

$$FA = \frac{M \cdot cp}{1.08 \cdot FV \cdot R}$$

$$= \frac{Q}{(T_1 - T_2)1.08 \cdot FV \cdot R}$$

$$t_2 = R\,(T_1 - T_2) + t_1$$

$$Z = E \cdot R = \frac{T_1 - T_2}{T_1 - t_1}$$

FOR R BELOW R = 1.0

$$FA = \frac{R \cdot M \cdot cp}{1.08 \cdot FV}$$

$$= \frac{R \cdot Q}{(T_1 - T_2)1.08 \cdot FV}$$

$$t_2 = \frac{(T_1 - T_2)}{R} + t_1$$

Air Cooler Sizing Chart / 4 Pass

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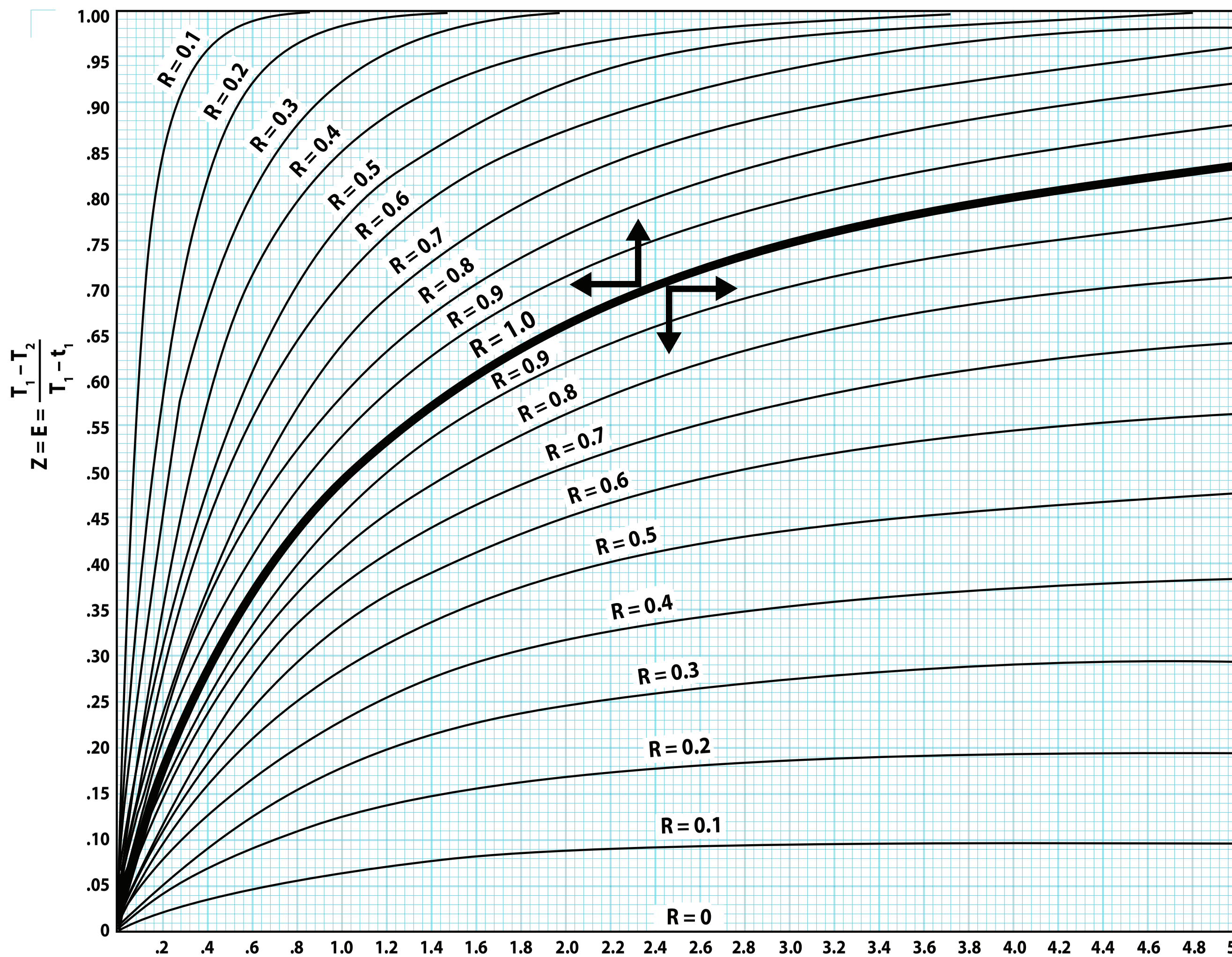
Figure 15

$$k = Ntu = \frac{n \cdot N \cdot a \cdot U}{1.08 \cdot FV} = \frac{n \cdot N \cdot a}{1.08 \cdot FV\,(r_{air} + r_i + r_f + r_m)}$$



FOUR (4) OR MORE PASSES

FOR R ABOVE R = 1.0

$$FA = \frac{M \cdot cp}{1.08 \cdot FV \cdot R}$$

$$= \frac{Q}{(T_1 - T_2)1.08 \cdot FV \cdot R}$$

$$t_2 = R\,(T_1 - T_2) + t_1$$

$$Z = E \cdot R = \frac{T_1 - T_2}{T_1 - t_1}$$

FOR R BELOW R = 1.0

$$FA = \frac{R \cdot M \cdot cp}{1.08 \cdot FV}$$

$$= \frac{R \cdot Q}{(T_1 - T_2)1.08 \cdot FV}$$

$$t_2 = \frac{(T_1 - T_2)}{R} + t_1$$

$$k = Ntu = \frac{n \cdot N \cdot a \cdot U}{1.08 \cdot FV} = \frac{n \cdot N \cdot a}{1.08 \cdot FV\,(r_{air} + r_i + r_f + r_m)}$$

Application of Design Method

For any ACHE service at the design stage, the given information includes the process terminal temperatures, the heat load, and the air ambient temperature as well as desirable tube dimensions. The Ntu approach to design determines the optimum values for the face area of the bundle and the airside outlet temperature. Using this data, a cooler design is selected which must be checked rigorously, but the selection most likely will be close to the final best design for a given service.

The value of Z can be calculated from the given data, and the overall heat transfer coefficient estimated from values shown in Table 2. This allows the determination of the number of tube rows and face velocity corresponding to the value of Z • 100/U in Table 1. The value of k or Ntu is calculated and, using the assumed number of tube passes, R is read using the appropriate curve in Figures 12 through 15. This value of R is applied in the appropriate equations to predict values of FA and the air outlet temperature. This design can be assumed to be accurate enough to be used for estimating purposes. An example of this process is shown in the sample problem on the following page.

This selection must be checked rigorously to create the final design. This is done by application of heat transfer and pressure drop correlations that have been developed empirically and confirmed by experimentation and observation of air cooler performance. Many of these correlations are generally known. Hudson Products applies criteria that have been developed over the past 40 years of experimentation and observation. The influence of the various parameters that affect both heat transfer and pressure drop are continually being updated by Hudson Products to assure the application of the latest techniques in the design of air-cooled heat exchangers.

TABLE 1

$Z \bullet \frac{100}{U}$	Rows	FV (ft/min)
0.4	4	650
0.5	5	600
0.7	6	550
0.8 to 1.0	8 to 10	400 to 450

TABLE 2

TYPICAL HEAT TRANSFER COEFFICIENTS FOR AIR-COOLED HEAT EXCHANGERS

Condensing service	U
Amine reactivator	100 – 120
Ammonia	105 – 125
Refrigerant 12	75 – 90
Heavy naphtha	70 – 90
Light gasoline	95
Light hydrocarbons	95 – 105
Light naphtha	80 – 100
Reactor effluent Platformers, Hydroformers, Rexformers	80 – 100
Steam (0 – 20 psig)	135 – 200

Gas cooling service	
Air or flue gas @ 50 psig (ΔP = 1 psi)	10
Air or flue gas @ 100 psig (ΔP = 2 psi)	20
Air or flue gas @ 100 psig (ΔP = 5 psi)	30
Ammonia reactor stream	90 – 110
Hydrocarbon gasses @ 15 – 50 psig (ΔP = 1 psi)	30 – 40
Hydrocarbon gasses @ 50 – 250 psig (ΔP = 3 psi)	50 – 60
Hydrocarbon gasses @ 250 – 1500 psig (ΔP = 5 psi)	70 – 90

Liquid cooling service

Engine jacket water	130 – 155
Fuel Oil	20 – 30
Hydroformer and Platformer liquids	85
Light gas oil	70 – 90
Light hydrocarbons	90 – 120
Light naphtha	90
Process Water	120 – 145
Residuum	10 – 20
Tar	5 – 10

Coefficients are based on outside bare tube surface for 1-inch OD tubes with 10 plain extruded aluminum fins per inch, 5/8 inch high, 21.2:1 surface ratio.

Sample Problem

Cool 273,000 lb/hr of light hydrocarbon liquid from 250°F to 150°F with 100°F ambient air, and with DPallowable = 5 psi, fouling = 0.001, elevation = sea level, and properties at the average tube-side temperature of 200°F as follows:

c_p = 0.55 Btu/lb • °F

k_i = 0.055 Btu/(hr • ft • °F)

μ = 0.51 centipoise = 1.234 lb/ft • hr

The specified tube material is 0.085 inch MW (0.093 inch AW) x 32 foot long carbon steel, and we select 1-inch OD tubes with 10 extruded fins per inch, 5/8 inch high, with 2.5 inch transverse tube pitch.

We calculate Q = 273,000 • 0.55 • (250 – 150) = 15,015,000 Btu/hr, and then use the Ntu Method to select an exchanger size to try.

$$Z = \left[\frac{250 - 150}{250 - 100}\right] \bullet \frac{100}{90} = 0.741$$

From Table 1, we see that proper number of tube rows is 6, and that the face velocity should be about 550 sfm. From Table 2, the overall heat transfer coefficient should be about 90 Btu/(hr • ft² • °F). We can then calculate k:

n = Tubes per row, per foot of width

$$= \frac{12}{\text{pitch}} = \frac{12}{2.5} = 4.8$$

$$a = \frac{\pi}{12} \bullet \text{OD} = 0.2618$$

$$k = \frac{n \bullet N \bullet a}{1.08 \bullet FV \bullet (1/U)}$$

$$= \frac{4.8 \bullet 6 \bullet 0.2618}{1.08 \bullet 550 \bullet (1/90)} = 1.1424$$

We will assume three tube-side passes, and find the heat capacity ratio R from Figure 14 to be 0.70 and above the R = 1.0 line, so the R term in FA is in the denominator. We can then calculate the face area:

$$FA = \frac{Q}{FV \bullet 1.08 \bullet R \bullet (T_1 - T_2)} = 360.8$$

Considering the given 32 foot long tubes,

$$\text{Width} = \frac{360.8}{32} = 11.28 \text{ ft}$$

$$\text{The tube count} = \frac{11.28 \text{ ft} \bullet 12 \text{ in/ft} \bullet 6 \text{ rows}}{2.5 \text{ inch tube pitch}} = 325$$

The width of the structural components including air seals adds about 6 inches, and we round the size up to the nearest standard size, namely 12 feet, with 336 tubes.

Fan Selection – Horsepower Requirements

The fan diameter must assure that the area occupied by the fan is at least 40 percent of the bundle face area. The fan diameter must be 6 inches less than the bundle width. Fan performance curves are used to select the optimum number of blades and pitch angle as well as the horsepower.

To calculate the required horsepower for the fan driver:

Motor Shaft Horsepower =

$$\frac{\text{Actual ft}^3\text{/min (at fan)} \bullet \text{Total Pressure Loss (inches water)}}{6356 \bullet \text{Fan (System) Efficiency} \bullet \text{Speed Reducer Efficiency}}$$

The actual volume at the fan is calculated by multiplying the standard volume of air (scfm) by the density of standard air (0.075 lb/ft^3) divided by the density of the air at the fan. From this relationship, it can be seen that the ratio of the fan horsepower required for a forced draft unit to that required for an induced draft unit is approximately equal to the ratio of the exit air density to the inlet air density, which is in turn equal to the ratio of absolute air temperatures $(t_1 + 460) / (t_2 + 460)$. The total pressure difference across the fan is equal to the sum of the velocity pressure for the selected fan diameter, the static pressure loss through the bundle, (which is determined from the equipment manufacturer's test data for a given fin type and tube spacing), and other losses in the aerodynamic system. Fan diameters are selected to give good air distribution and usually result in velocity pressures of approximately 0.1 inch of water.

The design of the fan, the air plenum chamber, and the fan housing, (in particular fan tip clearance), can materially affect system efficiency, which is always lower than on fan curves based on idealized wind tunnel tests. Industrial axial flow fans in properly designed ACHEs have fan (system) efficiencies of approximately 75%, based on total pressure. Poorly designed ACHEs may have system efficiencies as low as 40%. Speed reducers usually have about 95% mechanical efficiency. The value of driver output horsepower from the equation above must be divided by the motor efficiency to determine input power.

For estimating purposes, refer to Figure 16 to approximate the horsepower requirement. This chart plots bare tube surface divided by horsepower versus tube bundle depth for the normal range of velocities. Applying the above criteria to our sample problem, we determine that we must use two 10-foot diameter fans to have 40% of the bundle face area. Entering Figure 16, we find that for a 6-row bundle, the area/horsepower is between 68 and 92 square feet of bare tube surface. If we use an average value of 80, the horsepower requirement for each fan is $(336 \bullet .2618 \bullet 32) / (2 \bullet 80) = 17.5$ horsepower at maximum design ambient temperature. Power consumption must be calculated for the coldest expected ambient temperature, since at a fixed fan blade angle, fan horsepower consumption is inversely proportional to the absolute temperature. The power required for this minimum ambient temperature will set the required motor size.

III. PERFORMANCE CONTROL OF ACHEs

In addition to the fact that the process flow rate, composition, and inlet temperature of the fluid may vary from the design conditions, the ambient air temperature varies throughout a 24-hour day and from day to day. Since air coolers are designed for maximum conditions, some form of control is necessary when overcooling of the process fluid is detrimental, or when saving fan power is desired. Although control could be accomplished using by-passing of process fluid, this is rarely done, and the usual method is air flow control.

Unit Weight And Surface Per Unit Fan Horsepower As A Function Of Bundle Depth

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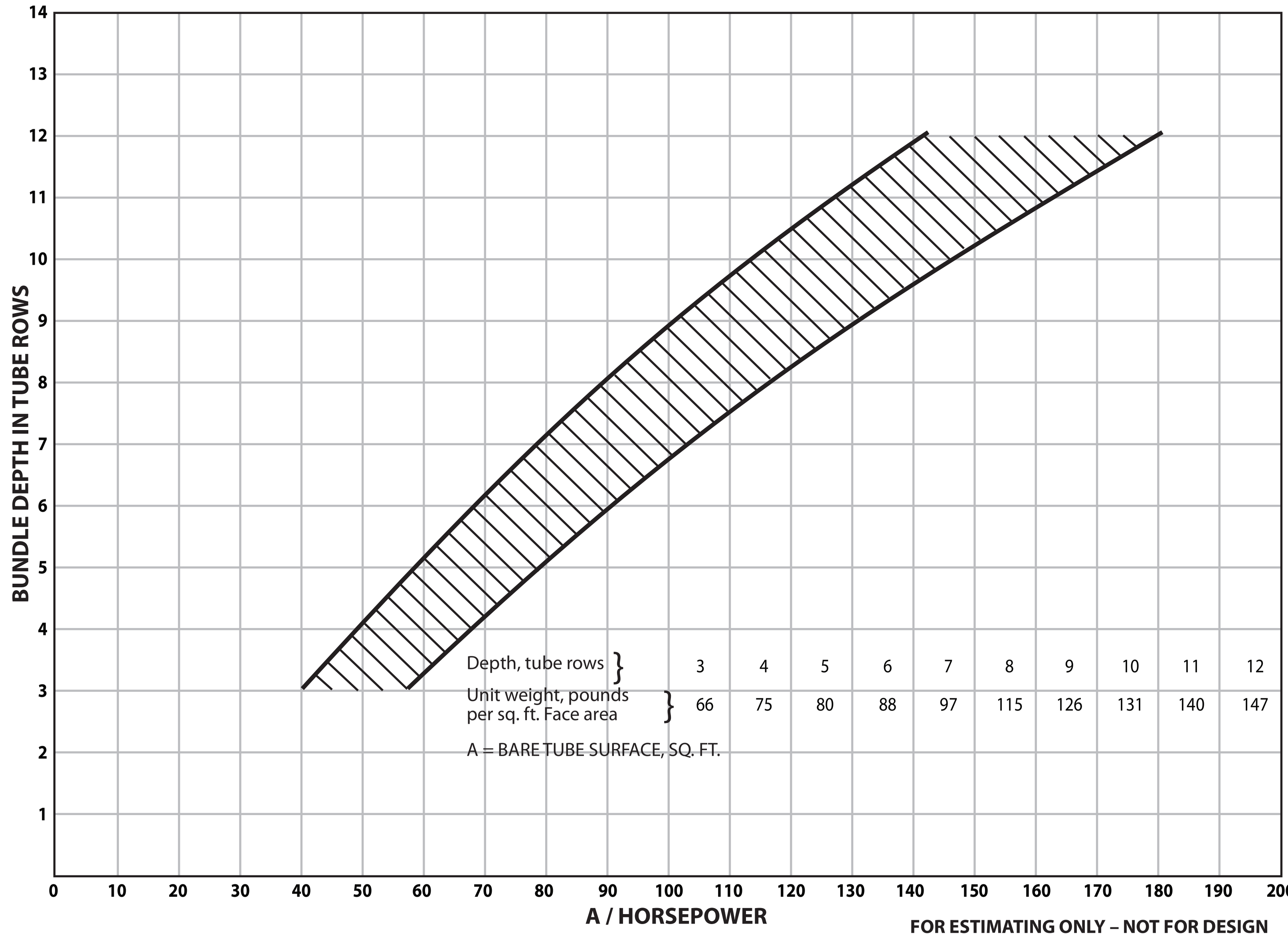
Figure 16



Varying Air Flow

Varying air flow can be accomplished by:
(See Figure 17)

1. Adjustable louvers on top of bundles.
2. Two-speed fan motors.
3. Fan shut-off in sequence for multifan units.
4. AUTO-VARIABLE® fans.
5. Variable frequency for fan motor control.

Louvers operate by creating an adjustable restriction to air flow and therefore do not save energy when air flow is reduced. In fact, louvers impose a permanent energy loss, even in the open position.

Two-speed motors, AUTO-VARIABLE fans, and variable frequency fan motor control do save power when air flow is reduced. In temperate climates, as much as 67% of the design power may be saved over the course of a year with AUTO-VARIABLE pitch fans. AUTO-VARIABLE hubs will thus pay back their additional cost in about one year or less.

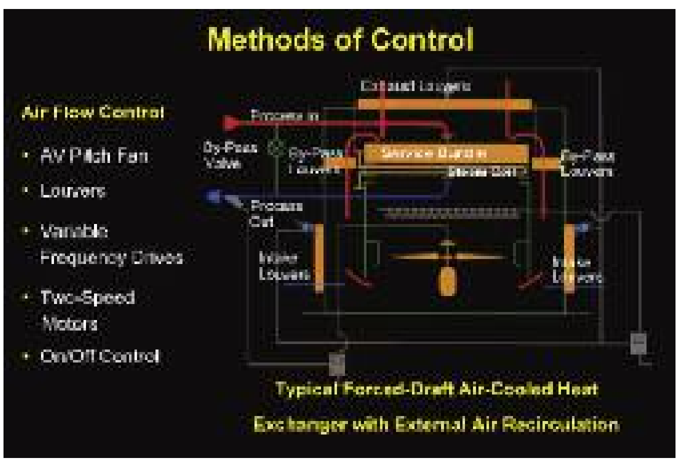


Figure 17

Both louvers and AUTO-VARIABLE fans may be operated automatically through an instrument that senses temperature or pressure in the outlet header. For extreme cases of temperature control, such as prevention of freezing in cold climates in winter, or prevention of solidification of high pour-point or high melting point materials, more sophisticated designs are available.

Extreme Case Controls

Extreme case controls include:

1. Internal Recirculation
By using one fixed-pitch fan blowing upward and one AUTO-VARIABLE pitch fan, which is capable of negative pitch and thus of blowing air downward, it is possible to temper the air to the coldest portion of the tubes and thus prevent freezing. Normally forced draft units have the negative pitch fan at the outlet end, while induced draft units have the positive pitch fan at the outlet end. In hot weather, both fans can blow upward.

2. External Recirculation
This is a more positive way of tempering coolant air, but is practical only with forced draft units. Hot exhaust air exits the bundle and enters a top plenum covered by a louver. When no circulation is required, the top louver is wide open, and the heated air exits through it. When the top louver is partially closed, some of the hot air is diverted to a duct, through which it flows downward and back into the fan intake, mixing with some cold ambient air. An averaging air temperature sensor below the bundle controls the amount of recirculated air, and thus the average air intake temperature, by varying the louver opening.

3. Co-current Flow
Four high pour-point streams, it is often advisable to ensure a high tube wall temperature by arranging the flow co-currently, so that the high inlet temperature process fluid is in contact with the coldest air and the low temperature outlet process fluid is in contact with the warmed air.

4. Auxiliary Heating Coils – Steam or Glycol
Heating coils are placed directly under bundles. Closing a louver on top of a bundle will allow the heating coil to warm the bundle or keep it warm in freezing weather, so that on start-up or shut-down the material in the bundle will not freeze or

solidify. Heating coils are also occasionally used to temper very cold air to the bundles while the fan is operating and the exhaust louver is open.

IV. NOISE CONTROL

In recent years concerns about industrial noise have grown. Since ACHEs were not originally one of the serious sources, it has only been after the abatement of the more serious contributors that attention has focused on ACHEs.

ACHE noise is mostly generated by fan blade vortex shedding and air turbulence. Other contributors are the speed reducer (high torque drives or gears) and the motor. The noise is generally broad band, except for occasional narrow band noise produced by the motor or speed reducer, or by interaction between these sources and the ACHE structure.

The evidence is that for efficient fans at moderate fan tip speeds, this noise is proportional to the third power of the fan blade tip speed and to the first power of the consumed fan horsepower. It is at present quite practical and usually economical to reduce the sound pressure level at 3 feet below an ACHE to 85 dB(A); but, below 80 dB(A), noise from the drives predominates and special measures must be taken.

V. DESIGN OF ACHEs FOR VISCOUS LIQUIDS

Film coefficients for laminar flow inside tubes are very low and of the same order of magnitude as film coefficients for air flowing over the outside of bare tubes. Therefore, there is generally no advantage in using fins on the air side to increase the overall heat transfer rate since the inside laminar flow coefficient will be controlling. Bare tube bundles with a large number of rows are usual.

For process fluids with outlet viscosities up to 20 centipoises, it is possible by using large diameter tubes and high velocities (up to 10 ft./sec) to achieve a Reynolds number at the outlet above the 2,000 critical Reynolds number, and to keep the flow in the transition region. However, this usually results in pressure drops of 30 to 100 psi. In view of the disadvantages of designing for laminar flow, this increased pressure drop is normally economically justifiable because the increase in the operating and capital cost of the pump is small compared with the decrease in the cost of the turbulent exchanger.

The biggest problem with laminar flow in tubes is that the flow is inherently unstable. The reasons for this can be demonstrated by a comparison of pressure drop and heat transfer coefficient for turbulent versus laminar flow, as functions of viscosity (μ) and mass velocity (G):

Flow Type	Delta P Function	Heat Transfer Function
Turbulent	$\mu^{0.2}$, $G^{1.8}$	$\mu^{-0.47}$, $G^{0.8}$
Laminar	$\mu^{1.0}$, $G^{1.0}$	$\mu^{0.0}$, $G^{0.33}$

In an air-cooled heat exchanger, because of imperfect air-side flow distribution due to wind, or because of multiple tube rows per pass, it is likely that the flow through some of the tubes in a given pass is cooled more than that through other tubes.

With turbulent flow, pressure drop is such a weak function of viscosity (0.2 power) and such a strong function of mass velocity (1.8 power), that the flow in the colder tubes must decrease only slightly in order for the pressure drop to be the same as that in the hotter tubes. Also, as the flow slows and the viscosity increases, the heat transfer coefficient drops significantly, (-0.47 power of viscosity, 0.8 power of G), so the over-cooling is self-correcting.

With laminar flow, pressure drop is a much stronger function of viscosity (1.0 power) and a much weaker function of mass velocity (1.0 power), so the flow in the colder tubes must decrease much more to compensate for the higher viscosity. Viscosity of heavy hydrocarbons is usually a very strong function of temperature, but with laminar flow, the heat transfer coefficient is independent of viscosity, and only a weak function of mass velocity (0.33 power), so the self-correction of turbulent flow is absent.

The result is that many of the tubes become virtually plugged, and a few tubes carry most of the flow. Stability is ultimately achieved in the high flow tubes as a result of high mass velocity and increased turbulence, but because so many tubes carry little flow and contribute little cooling, a concurrent result is high pressure drop and low performance. The point at which stability is reached depends on the steepness of the viscosity versus temperature curve. Fluids with high pour points may completely plug most of an exchanger.

This problem can sometimes be avoided by designing deep bundles to improve air flow distribution. Bundles should have no more than one row per pass and should preferably have at least two passes per row, so that the fluid will be mixed between passes.

When fluid has both a high viscosity and a high pour point, long cooling ranges should be separated into stages. The first exchanger should be designed for turbulent flow, with the outlet temperature high enough to ensure an outlet Reynolds number above 2,000 even with reduced flow. The lower cooling range can be accomplished in a serpentine coil (a coil consisting of tubes or pipes connected by 180° return bends, with a single tube per pass). The low temperature serpentine coil should, of course, be protected from freezing by external warm air recirculation ducts.

Closed loop tempered water systems are often more economical, and are just as effective as a serpentine coil. A shell and tube heat exchanger cools the viscous liquid over its low temperature range on the shell side. Inhibited water is recirculated between the tube side of the shell and tube and an ACHE, where the heat is exhausted to the atmosphere.

For viscous fluids which are responsibly clean, such as lube oil, it is possible to increase the tube side coefficient between four- and tenfold, with no increase in pressure drop, by inserting turbulence promoters and designing for a lower velocity. It is then advantageous to use external fins to increase the air-side coefficient also. In addition to the increase in heat transfer coefficient, turbulence promoters have the great advantage that the pressure drop is proportional to the 1.3 power of mass velocity, and only to the 0.5 power of viscosity, so that non-isothermal flow are much more stable. The simplest and probably the most cost-effective promoters are the swirl strips, a flat strip twisted into a helix.

VI. COST

The approximate purchase price may be determined from Figure 18, which gives the price per square foot of bare tube surface as a function of the total bare tube surface and the number of tube rows. The prices indicated are FOB factory, and do not include freight or export crating charges. The prices are based on 1-inch OD X 12 BWG X 32 foot long steel tubes with extruded aluminum fins, fabricated steel headers with steel shoulder plugs, 100 psig design pressure, TEFC motors, and HTD drives. Price multiplication factors are included for different tube materials.

It can be seen from these curves that the price per square foot varies little for installations in excess of 7,000 square feet of bare tube surface. It is also evident that the reduction in unit price as a function of the number of tube rows becomes progressively less as the number of rows increases.

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Unit Price As A Function Of Total Surface And Bundle Depth

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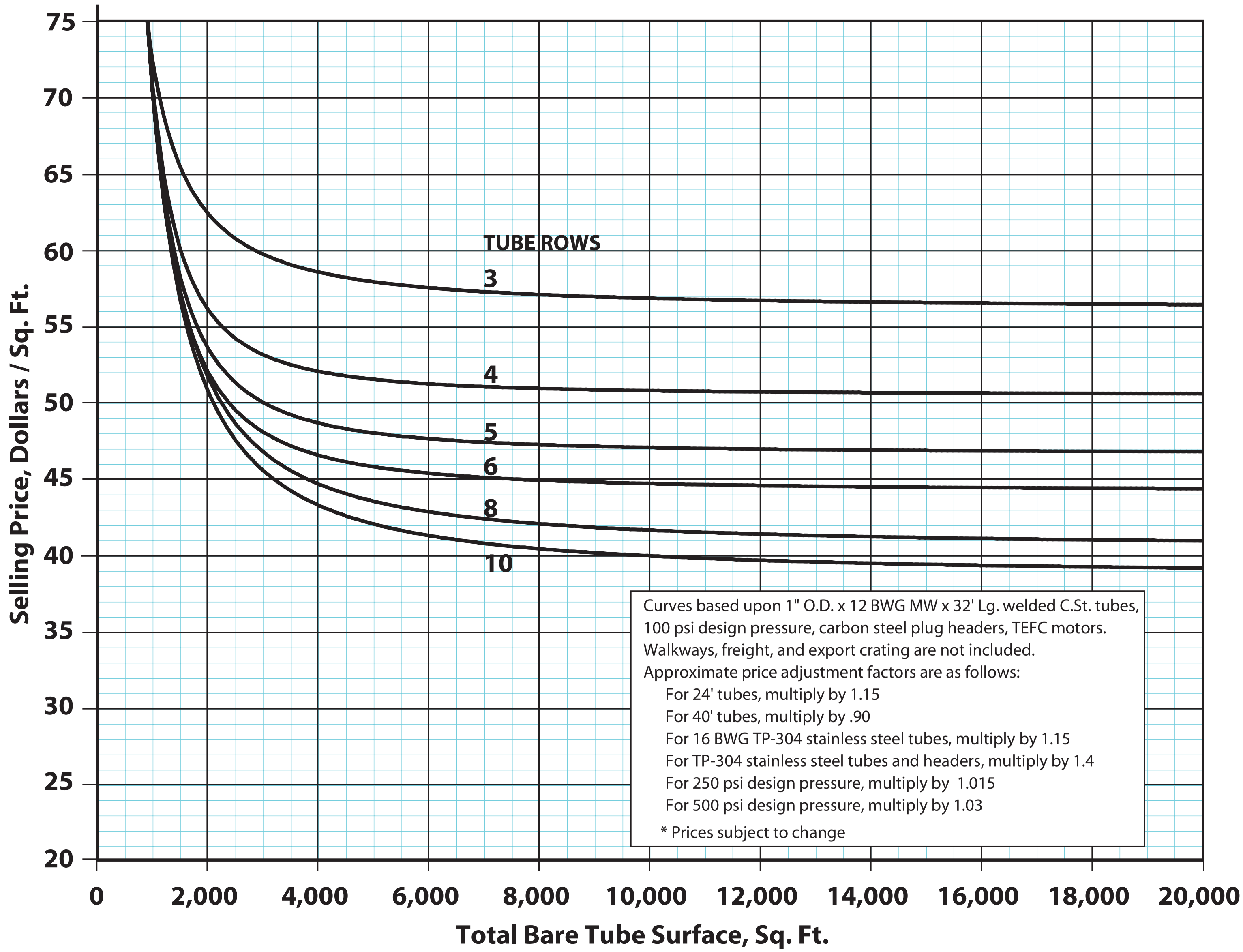
Figure 18



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