

EXXON EXXON ENGINEERING	HEAT EXCHANGE EQUIPMENT AIR-COOLED HEAT EXCHANGERS PROPRIETARY INFORMATION - For Authorized Company Use Only	DESIGN PRACTICES	
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Revision Memo

12/98 General revisions and updated information throughout the section.



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SCOPE

This section describes air-cooled heat exchangers (ACHEs) and provides methods for estimating air-cooled heat exchanger surface, plot area, and fan power requirements. It is intended that the methods be used for determining the economics of ACHE vs. water coolers, making estimates, or for checking contractor's bids, etc. Both a short cut and a detailed procedure are provided. The choice between the two methods should be determined by the final use of the results. Also presented are design considerations and guidelines for determining an optimum air-cooled exchanger configuration.

The application and selection of air-cooled exchangers are discussed in [Section IX-A](#). Mechanical requirements are covered in [IP 6-4-1](#) which supplements API Standard 661: *Air-Cooled Heat Exchangers for General Refinery Services*.

For detailed calculations, a number of computer programs are available: (1) HTRI's ACE-2 program is the recommended modeling tool and is available in Hextran, (2) Program 3195, which uses AC-2 Program developed by PFR Engineering, (3) The HEXTRAN Program provides an ACE module for the design and rating of air-cooled exchangers, and (4) The ACOL program developed by HTFS. In addition, HTRI provides a number of reports (AC-1 through AC-9 & ACD 1, 2.1, 2.2) on their research activities in this area. Similar reports (DR-39, -54 and -59) are also given by HTFS. [Section IX-K](#) provides additional information on these programs.

REFERENCES

DESIGN PRACTICES

Section II	Design Temperature, Pressure & Flange Ratings
Section X	Pumps
Section XII	Instrumentation
Section XV	Safety in Plant Design
Section XIV-D	Two Phase Flow
Section IX-A	Exchanger Types and Applications
Section IX-C	Design Considerations for Shell and Tube Exchangers

INTERNATIONAL PRACTICE

IP 6-4-1 ,	Air-Cooled Heat Exchangers (Supplements API-661, Air-Cooled Heat Exchangers for General Refinery Services)
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OTHER LITERATURE

1. Robinson, K.K. and Briggs, D.E., *Pressure Drop of Air Flowing Across Triangular Pitch Banks of Tubes*, A.I.Ch.E. Preprint 20, Eighth National Heat Transfer Conference A.I.Ch.E. - A.S.M.E., Los Angeles, California, (August, 1965).
2. Kayes, W.M. and London, A.L., *Compact Heat Exchangers*, National Press, Palo Alto, California, (1955).
3. Lapin, A. and Schurig, W.F., *Heat Transfer Coefficients for Finned Exchangers*, Ind. & Eng. Chem. 51: 941-4, (August, 1959).
4. *Exxon Refinery Construction Materials Manual (RCMM)*, ER & E Manual No. EETD 828.
5. Simulation Sciences, Inc., Hextran-HTRI Input Guide.
6. Hinrichsen, M.P., *Wet-Surface ACHE and Evaporative Coolers*, Exxon Engineering Report EE.16E.94, January, 1994.

DEFINITIONS

Tube Bundle - Assembly of headers, tubes, and frame. Also referred to as a "Section."

Bay - One or more tube bundles served by one or more fans complete with structure, plenum and other attendant equipment.

Unit - One or more bays for an individual service.

Bank - One or more bays including one or more units arranged in a continuous structure.

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AIR-COOLED HEAT EXCHANGER TYPES

Air-cooled exchangers (ACHES) are composed of rectangular bundles containing several rows of tubes on a triangular pitch (see **Figures 1 and 2**). Heat transfer is generally countercurrent; the hot fluid entering at the top of the bundle and air flowing vertically upward through the bundle. Bundles may be fabricated in widths up to 16 ft (4.9 m) and depths up to 10 rows. Usually the maximum dimensions are dictated by shipping requirements. Field-erected bays could be larger and the size is limited by tube length, fan diameter or available plot area. The most common tube length is 30 ft (9 m) although bundles are available in lengths from 8 to 60 ft (2.4 to 18.3 m). Bundles may be placed in parallel, and several small services may be combined in one bay.

FORCED & INDUCED DRAFT

The two most common types of ACHE are: (1) forced-draft type, where air is pushed across the tube bundle, and (2) induced-draft type, where air is pulled through the bundle (see **Figure 2**). The advantages of each type are listed below.

Advantages of Forced Draft

1. Generally requires less fan power than induced draft for an air temperature rise greater than 50°F (28°C), because the fan is pumping denser air.
2. Adaptable for winterization, pour-point, and hot air recirculation schemes.
3. Mechanical equipment is more readily accessible for maintenance, and the fan shaft does not penetrate the tube bundle.
4. Less structural support is required, and capital cost is lower.
5. No mechanical equipment is exposed to hot exhaust air. Induced draft units have mechanical equipment exposed to hot outlet air, which limits their outlet air temperature to approximately 300°F (149°C) or less depending on the equipment type.

Advantages of Induced Draft

1. Less hot air recirculation as exhaust air velocity is about 2 1/2 times that of forced draft. Induced draft units have a transition plenum, which provides more uniform air distribution across the bundle (see **Figure 2**).
2. Offers bundle protection from adverse weather (rain, hail, snow, etc.). Also, shields the bundle from solar heating and rain quenching, which can result in sudden temperature changes and process upsets.
3. Better suited for cases with close approach temperatures between air inlet and fluid outlet.
4. Will transfer more heat by natural convection with fans off because of the stack effect.

Recommendations

1. Induced-draft units should be used whenever hot-air recirculation must be avoided.
2. Forced-draft units should be used whenever the design requires pour-point protection, or winterization. However, air recirculation during summer must be accounted for in sizing the fans.

A-FRAMES

To accommodate plot area restrictions an A-Frame type bundle may be used (see **Figure 2**). This design is adaptable for roof-mounted units and is field erected. The plot area requirement of the A-Frame is about half that for a conventional flat bundle. The design can be oriented as an A-Frame, a V-Frame, or a flower petal. A major disadvantage is that the A-Frame design is highly susceptible to recirculation problems. Therefore, unless the possibility of a recirculation problem is minimal, it is recommended that a flat bundle arrangement be used.

A-Frames are almost always single pass designs and are typically only used in low-pressure steam condensing services (i.e., surface condensers). A-frames are not used for condensing hydrocarbons because of the lack of experience in condensing hydrocarbons in inclined tubes.

It is estimated that A-Frames cost 30-40% more than the corresponding horizontal exchanger. The cost of erection can be 2 to 3 times that of horizontal bundles. Typical A-Frames do not allow for mechanical cleaning inside the tube.



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AIR-COOLED HEAT EXCHANGER TYPES (Cont)

WET SURFACE AIR-COOLED HEAT EXCHANGER (WSACHE)

Wet Surface Air-Cooled Heat Exchangers combine the technology of the ACHE and the Cooling Tower into one piece of equipment. WSACHES should be considered when it is desired to cool the process fluid close to the ambient temperature and when there is limited amount of cooling water available. Unlike typical ACHE, WSACHES have the advantage of cooling the process stream against the ambient wet-bulb temperature. Because the latent heat of water is very high, much less cooling tower water is required to operate a WSACHES than a Shell & Tube exchanger that operates under sensible heating. The larger airside heat transfer coefficients and increased temperature differences of WSACHES result in exchangers that are smaller than the equivalent ACHEs. An example and description of a typical type of WSACHE is shown in **Figure 3**.

NATURAL DRAFT

Natural Draft Air-Cooled Heat Exchangers require no fan to provide air-flow. A stack on top of the bundle prevents the hot bundle exit air from mixing with the cooler ambient air. The air is driven by the differences in densities between the ambient air and the air at the bundle exit. Natural Draft ACHE may be considered for applications where there is a limited electrical power supply, where noise level is a dominating issue, and for services where fan shutdown would result in critical conditions. ND ACHEs should also be considered to reduce total flare load associated with power failure and whenever the design process outlet temperature is above 350°F (177°C).

Note that the design of A-Frames, WSACHE and Natural Draft ACHE require special design considerations that are not discussed in this Design Practice.

AIR-COOLED HEAT EXCHANGER COMPONENTS

FANS

Usual ACHE fan sizes range from 4 to 16 ft (1.2 to 4.9 m) in diameter. This gives a fair amount of leeway in the arrangement of bundles. The maximum size variable-pitch fan is 16 ft (4.9 m) in diameter and the minimum size is 5 ft (1.5 m). It is common in practice to have two fans in each bay to ensure adequate temporary cooling in the event of a failure of one of the fans. Also it is good practice to keep the projected face area of the fans a minimum of 40% of the face area of the bundles in the bay supplied by the fans.

TUBES

Tubes for air-cooled heat exchangers may be plain or may contain fins. The fin density varies between 7 and 11 fins/in. (276 and 433 fins/m) and fin height varies from 5/16 to 1 in. (8 to 25 mm). Typically the fin material is different than the tube material with aluminum fins commonly used. The fins may be extruded from a tight fitting sleeve that fits over the tube, embedded into the tube, wrapped into spiral grooves cut into the tube, or tension wound onto the tube (see **Figure 4**). The extruded finned tube affords the best protection of the liner tube from atmospheric corrosion.

The maximum size finned tube that is available is 2 in. (51 mm). The minimum acceptable tube diameter (**IP 6-4-1**) is 1 in. (25 mm). The same tube-side velocity limitations, which apply to shell-and-tube exchangers (**Section IX-C**), apply for ACHEs. Tube material selection and thickness requirements should be established using the Refinery Construction Material Manual (**Reference 4**) or through consultation with a materials specialist. Fin types are limited to the following maximum design temperatures:

FIN TYPE	MAXIMUM DESIGN TEMPERATURE °F (°C)
Embedded	750 (400)
Extruded	500 (260)
Footed Tension Wound	300 (150)
Edge Tension Wound ⁽¹⁾	250 (120)

Note:

(1) Cannot be used for steam condensing service.

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AIR-COOLED HEAT EXCHANGER COMPONENTS (Cont)

The maximum design temperatures in the chart above are based on the expansion of the fin relative to the tube. Since the fin is at a lower temperature than the tube, the tube will want to expand more than the fin. The interference stress between the tube and fin may cause the fin to become plastically deformed. Therefore, when the ACHE is turned off, a gap may exist between the tube outside diameter and the fin, resulting in decreased heat transfer.

The fins may be serrated or plain. Serrated fins, as shown in **Figure 5**, are formed from extruded fins and provide increased air-side heat transfer coefficients and pressure drops for a given air-side velocity. Serrated fins foul more easily than plain fins.

Several European vendors claim to have used elliptical tubes successfully for many years. These tubes have increased resistance to water freezing for steam condensing in cold environments. In many applications a single row of elliptical tubes may be used instead of several rows of conventional tubes. The single tube row ensures that all tubes have almost equal flow.

For very viscous flows it is sometimes advantageous to use turbulence promoters, such as the Twisted Tape Insert, to increase the heat transfer coefficients of the process side fluid. Increasing the tubeside heat transfer coefficient makes adding air-side fins beneficial. Turbulence promoters should only be used in clean services. Although it is less common, internal fins may be used to increase the tubeside heat transfer coefficient. Turbulence promoters and tubeside enhancements are discussed in **Section IX-A Exchanger Type and Applications**.

Selection of fin dimensions depends upon the specific service. Some general rules are:

1. If the overall heat transfer coefficient (referred to as bare tube area) is greater than 20 Btu/hr-ft²-°F (114 W/m²-K) or if the fluid viscosity is less than 10 cP (0.010 Pa•s), the higher fins (0.625 in. or 16 mm) are used.
2. If the overall coefficient is between 15 and 20 Btu/hr-ft²-°F (85 and 114 W/m²-K), or if the fluid viscosity is in the range of 10 to 25 cP (0.010 to 0.025 Pa•s), intermediate size fins (0.3125 in. or 8 mm) are used.
3. If the overall coefficient is below 15 Btu/hr-ft²-°F (85 W/m²-K), or if the fluid viscosity is greater than 25 cP (0.025 Pa•s), no fins are used.

TUBE SUPPORTS

The maximum unsupported tube span for ACHes is 6 ft (1.8 m). Various types of tube supports are used by individual vendors. Most tube supports consist of a narrow piece of metal that has been bent to accommodate the triangular pitch of an ACHE. The metal strips go either between or around the tubes, allowing the tubes to support each other.

HEADERS

Examples of the common header types used on air-cooled exchangers are shown in **Figure 6**. The majority of units in service use the plug header - a box design using a shoulder plug opposite each tube end to permit access to each tube hole for cleaning and rolling. A second type, the coverplate header, replaces the tube plugs and plug sheet with a flanged coverplate. Coverplate headers are used for moderate pressure [up to 300 psi (2.1 MPa)] and high fouling services. A modification of the coverplate design is the bonnet type, using a half pipe end construction instead of a flat plate with the closure joint at the tube-sheet end. For very high-pressure service [up to 6000 psi (42 MPa)] a manifold type header is used.

A horizontally split header, as shown in **Figure 6**, may be required to accommodate differential tube expansion in services having high fluid temperature differences per pass. The horizontal split allows the upper and lower sections to expand independently of each other.

NOZZLES

The typical inside width of a box header is 4 in. (102 mm). To accommodate large nozzles, the nozzle has to be flattened to an elliptical shape at the end that gets welded to the header box. Most nozzles have inside diameters less than 12 in. (305 mm). Although it will vary for individual applications, in general the distance between nozzles of a header should not exceed 4 ft (1.2 m). Large distances between individual nozzles will result in unequal distribution of the flow.

PLENUM

The plenum is an enclosure that provides a smooth flow of air between the fan and the bundle. There are two types of plenums, transition and box (see **Figure 2**). Transition plenums provide a more uniform distribution of air over the bundle than the box plenum. Transition plenums are rarely used for forced draft units because of the structural difficulties associated with hanging the machinery mounts.

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AIR-COOLED HEAT EXCHANGER COMPONENTS (Cont)

SPEED REDUCERS

Speed reducers allow the fans shaft to rotate at a speed lower than the speed at which the motor's shaft rotates. The three most common types of speed reducers are:

1. V-Belts: Cheapest and least reliable reducer. May be two or three separated bands or one band up to 30 HP (22 kW).
2. Cog Belt: Most common. Has almost no slippage, hence has higher efficiency. More noisy than V-belt. Can deliver up to 50 HP (37 kW).
3. Right Angle Gear Drive: Used for HP > 50 (37 kW). Needs more maintenance (i.e., check and add grease).

CONTROLS

Several methods for controlling air-cooled heat exchangers are available. Control can be achieved on either the process side or the air side, or both. In brief, the methods available are:

PROCESS SIDE

1. Stepwise control by use of parallel bundle arrangement where bundles can be taken out of service.
2. Process fluid bypass (automatic or manual) for single-phase flow only.

AIR SIDE

1. **Multiple fans:** In a multiple fan installation, the fans can be arranged to allow them to be turned on or off individually. When less cooling is required, some of the fans can be turned off either manually or automatically. Typically, this is the control used on product coolers. It has the advantage of quick response and power cost savings when the motors are off. Its disadvantage is that it only provides step control, and therefore close control cannot be obtained. A multiple fan installation should not be used as a means of control for condensing services.
2. **Variable speed:** Air flow can be increased or decreased by varying the speed of the fans. This can be done by:
 - a. **Variable-frequency drive** - controls the fan and motor speed by varying the electric frequency. Where close control of process temperature [$\sim 2^\circ\text{F}$ (1°C)] is required, variable frequency drives (VFDs) may be specified. Although VFDs are relatively expensive compared to a fixed speed motor, the difference in system cost is less severe when compared to the additional cost of controls (i.e., louvers) that are needed if a VFD is not used.
 - b. **Steam turbine drive** with variable speed control - where low-cost steam is available and if the unit is located in a Division 1 explosion-proof area.
 - c. **Fluid drive** with a fixed-speed motor.
 - d. **Hydraulic drive**, where the hydraulic motors operate from a central source of hydraulic fluid. Requires maintaining additional equipment, such as seals.
 - e. **Two-speed motors** - provide better control than the multiple fan, on-off control, but it is still a "step type" operation.

All variable speed drives need to be tested throughout the full operating range of the system.
3. **Louvers (shutters) mounted on top of the unit:** Three types of louvers are available - parallel-blade, opposed-blade and progressive-action louvers. Of the three, the opposed blades are the most attractive in that they provide better air flow control than parallel louvers and do not suffer from linkage problems as much as progressive-action louvers. The cost of opposed-blade louvers is slightly more than the parallel-blade louvers but somewhat less than the cost for progressive-action louvers. In addition, the opposed blades allow the air discharge to be vertical, which should help combat air recirculation. Heavy-duty extruded aluminum louver blades should be used for their added strength while louver frames should be steel. Louvers can be automatically or manually controlled. Auto controls add from 10% to 30% to the investment cost.
4. **Louvered recirculation systems:** In addition to top-mounted louvers, an entire bay can be enclosed with a louver-duct system if process requirements dictate a necessity for warm-air recirculation (see [Figure 7](#)).
5. **Variable-pitch fans (automatic or manual):** Except for fans under 5 ft (1.5 m) diameter, all fans have adjustable pitch blades. Where close control of process temperature [$\sim 2^\circ\text{F}$ (1°C)] is required, autovaryable-pitch fans may be specified. It can be specified as 100% or a fraction of the quantity, such as 50%. The quick response-control characteristics for the A/V fans are continuously effective from minimum to maximum flow because only the required amount of air is used. This results in power saving. However, there are some negative feedbacks from users; careful selection of reliable vendors is required. Although A/V fans cost around three times more than manually adjustable pitch fans, the difference in system cost is less severe when compared to the additional cost of controls that is needed if an autovaryable pitch fan is not used.

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CONTROLS (Cont)

Selection of a control system requires consideration of many variables. For example, a major advantage of louvers is that they provide bundle protection for forced-draft units while allowing moderate control. A major disadvantage of louvers is that control of the process outlet temperature within 5°F (2.8°C) cannot be obtained because of poor control characteristics in the vicinity of the fully opened and fully closed positions. Along with being plagued by linkage problems, louvers consume more fan power than VFDs or autovvariable-pitch fans, because the air-flow rate is controlled by adding additional resistance to the system. Close control of the process outlet temperature [within ~ 2°F (1°C)] can be obtained with autovvariable-pitch fans or variable frequency drives. Sometimes both louvers and autovvariable-pitch fans are used for a critical service. The final selection of a control system will depend on the specific location and service, and whether continuous or seasonal control is desired.

AIR-COOLED HEAT EXCHANGER LAYOUT

Some general design guidelines covering air-cooler layout are as follows:

PIPING LAYOUT

Preliminary or proposed piping layout might have an influence on determining ACHE location, inlet/outlet header locations and odd/even tube pass requirement. Early familiarity of layout requirement will be helpful in ACHE design.

WIND DIRECTION ORIENTATION

Since warm-air recirculation can reduce ACHE performance, care should be taken in selecting the location and orientation of ACHEs with respect to the predominant wind direction. ACHE banks should be aligned with the tube length parallel to the predominant wind. In the case of a single bay or bank, this retards over-the-side warm-air recirculation. Often plot space requirements dictate the piperack as the site for ACHE location. This removes much of the flexibility with respect to orientation of the wind. Units with the "coldest" process outlet temperature requirement should be located upwind. They are usually the most susceptible to upset from warm-air recirculation and usually emit cooler air outlet temperatures. For high wind areas wind skirts are used especially if a furnace is located upstream.

ELEVATION

Process considerations may set the elevation of air-cooled exchangers. Frequently, the location of the distillate drum of a fractionating column will set the elevation of the overhead condenser. If process considerations do not dictate, the elevation should be sufficient to provide a peripheral area around the base of the unit at least large enough to provide a peripheral inlet air velocity no greater than 600 ft/min (3.0 m/s). Any obstructions to inlet air flow will reduce the fan efficiency. Adjacent ACHE bays and banks should be located at equal elevations to prevent the hot air of the lower ACHE from entering the feed air of the higher ACHE. If adjacent ACHEs cannot be located at equal elevations, then wind fences or skirts should be applied to allow for equal hot air discharge elevations.

ORIENTATION RELATIVE TO OTHER ACHEs

To minimize over the side recirculation, bays within a unit or bank installed on a piperack should be placed as close to each other as possible as set by the existing pipe supports. If ACHEs are to be installed at grade then adjacent bays, banks, or units should be as close to each other as possible (within several in.) or at least two tube lengths apart. The minimum distances should comply with [Section XV-G, Equipment Spacing](#).

COMBINED SERVICES

Usually it is economically advantageous to combine small bundle services in a common bay. In doing so, care must be taken in locating each bundle because of air maldistribution in the plenum chamber. Critical service bundles and/or the smallest bundles should be situated in the center of the unit where the air distribution is good. Consideration should be given to oversurfacing the bundles located on the extremities, which may be subject to air starvation. Bundles should never straddle bays. Also, if louver control is specified, independent louver systems should be installed to obtain individual control of each service.

DESIGN CONSIDERATIONS

ACHE selection for a given service is based on a number of considerations. The most important of these are:

1. Economics of surface vs. fan power requirements (cost of electricity).
2. Process considerations such as allowable pressure drop, fluid characteristics (pour-point, viscosity, etc.), and weather conditions - summer recirculation or winterization problems.

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DESIGN CONSIDERATIONS (Cont)

- Fan type, size, static pressure drop limitations (maximum bundle depth for a given air requirement), and noise requirements.
- Plot area available, and pre-assembled or field-erected unit and erection cost.
- Mechanical considerations such as thermal expansion, structural supports, and maintenance requirements.
- Standardization of tubes, spare parts, etc.

SURFACE VS. FAN POWER

The economics of surface vs. fan power are fairly straight-forward. The significant variables in this regard are the level of cooling and the overall heat transfer rate. These variables determine the relative quantity of air required for a given heat duty. If the air quantity per unit heat load is comparatively low, the exchanger may consist of more tube rows, such that the air is pumped through with reasonable power consumption. However, if the air requirement is relatively high per unit heat load, the surface must be provided in less depth (fewer tube rows) and spread over a greater plan area. In the first case, the power is consumed in pumping a small quantity of air against a high static pressure, whereas in the latter case, a large air quantity is pumped against a lower static pressure. The maximum recommended static pressure drop across a fan for an air-cooled exchanger is 0.7 in. H₂O (0.175 kPa). Most fans, however, are designed to operate closer to 0.5 in. H₂O (0.125 kPa) static pressure drop. Depending on the fan, exceeding a certain static pressure drop will result in fan instability and a decrease in fan efficiency. The amount of pressure drop allowed before reaching instability increases with increasing fan tip speed. For mechanical and noise reasons the tip speed usually will not exceed 10,000 FPM.

In general, the longer the tubes and the greater the number of tube rows, the less expensive the surface on a square ft (m) basis.

PROCESS CONSIDERATIONS

Process considerations such as allowable pressure drop and fluid properties affect the number of tube passes per bundle and the bundle arrangement (series or parallel). Any reasonable number of tube passes may be obtained in an ACHE bundle and the same tubeside velocity limitations listed in [Section IX-C](#) for shell-and-tube exchangers apply. Besides the pressure drop requirement, the piping layout should also be considered in determining if an odd or even number of tube passes is required. There are, however, certain process considerations unique to air-cooled exchangers, which must be accounted for in the overall design of the unit:

Flow Distribution - If more than two bays are required, then an even number of bays is preferred. When uniform distribution of two-phase flow to more than one bundle is critical (such as when water wash must be evenly distributed), consider a tube arrangement of one row per pass. Also, upstream piping must be such that even distribution is provided to each bundle. (Refer to [Section XIV-D](#), *Two Phase Flow*.)

A maximum of one row per pass should be used when the process fluid is operating in the laminar flow regime. For laminar flow the tubeside pressure drop is a strong function of viscosity ($\mu^{1.0}$) compared to turbulent flow where the viscosity is to the power 0.2. If the design has more than one row per pass, the rows closer to the bottom will be colder and the viscosity will increase. This will result in more flow in the upper tubes than in the lower ones.

Design Air Inlet Temperature - Selecting a maximum design air inlet temperature is of primary importance. The maximum air inlet temperature for design should be a temperature high enough such that over the last five years only 5% of the ambient temperatures exceeded that temperature. To account for possible recirculation, a minimum of 5°F (3°C) should then be added to this value. In the case of a critical service, it may be advantageous to estimate maximum temperatures for the hottest months of the year and add 5°F (3°C) or more for recirculation. Critical services are defined as services where tower or process operation would be difficult to operate if the air temperature exceeded the design temperature for more than two hours. Similarly, a minimum design air inlet temperature, necessary for fan driver sizing and for determining possible winterization or pour-point services, should be selected. Maximum and minimum air inlet temperatures for various refinery locations are shown in [Table 1](#).

Freeze Protection - Winterization Services - The basic problem in this type of service is to prevent the tube wall temperature from going below the freeze/pour point of the process fluid. This may occur at low flow rates and/or low ambient air temperatures. The method used for preventing the fluid from reaching its freeze/pour point depends on the severity of the condition. Regardless of the severity, all air-cooled exchangers with freeze protection or winterization concerns should be forced draft units. For mild cases, such as where the ambient temperature goes only slightly below the freeze/pour point of the process fluid, the following options may be considered:

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DESIGN CONSIDERATIONS (Cont)

1. Steam coils may be used to increase the inlet air temperature. Steam coils generally consist of one row of tubes at twice the pitch of the tubes in the ACHE bundle. The steam coil goes between the fan and the bundle, and the coil should be sloped to help prevent freezing of the water condensate. As discussed in the *Controls* section, the type of air-flow control that is chosen should be based on the amount of control desired. Note that when using steam coils, louvers may be better than variable speed motors or A/V fans because louvers decrease the air flow rate throughout the entire bay. This can also be accomplished by using only A/V fans or only VFD drives.
2. When the process fluid has a high viscosity and is not condensing, a second alternative is to use cocurrent flow. Co-current flow puts the high inlet temperature process fluid in contact with the coldest air and the low temperature process outlet in contact with the warmest air. This causes in the wall temperature to be higher at the cold end of the exchanger. This method cannot be used when a temperature cross is required.
3. When the process fluid has a high viscosity and is not condensing, a third alternative is to design with no airside fins. Bare tubes provide a higher wall temperature for a given inside heat transfer coefficient. The total cost is not affected greatly because the poor inside coefficient negates the efficiency of finned tubing. However, sometimes it may be necessary to use low fin tubes to obtain a flow arrangement that provides a sufficient pressure drop.

For more severe cases where the ambient temperature may be further below the freeze/pour point of the process fluid, the following options may be considered:

1. Any combination of 1, 2, 3 from the above will provide higher wall temperatures.
2. Internal recirculation, commonly known as a Push-Pull, has one fan in a bay pushing air and A/V fan pulling the heated air over the bundle. As a minimum, top louvers must be supplied to prevent the heated air from leaving the system and allowing it to be pulled though by the A/V fan operating at a negative blade angle. Once the air is pulled through by the A/V fan, it mixes with the cold ambient air and is sucked in by the fan pushing the air. For better control, horizontal louvers can be added to the bottom sides or ends of the ACHE bay to control the amount of mixing that occurs between the ambient air and the heated air. The louvers control the temperature of the air entering the fan that is pushing the air, and the A/V fan is adjusted based on the process outlet temperature.

For the most severe cases, an external recirculation scheme should be used. A typical external recirculation scheme is shown in **Figure 7**. As the exhaust louvers close, the side bypass louvers open and allow the warm exhaust air to recirculate over the side of the bundle. The simultaneous closing of the intake louvers seals off the unit which then operates on a closed warm-air system. The ambient temperature will determine the degree to which the intake and exhaust louvers close to maintain a predetermined plenum mixture temperature. For all cases, depending on the ambient air temperature, it may be necessary to steam trace and/or insulate all exposed headers.

Hot-Air Recirculation - Hot-air recirculation is the phenomenon wherein heated effluent air recirculates back into the cool inlet air stream, mixes with it, and thereby raises the inlet air temperature. This can occur around a single unit or between the air streams of adjacent units. Problems associated with hot-air recirculation are the direct effect of poor exchanger design and location. Minimum allowable distances between air-coolers and other process equipment are given in **Section XV-G, Figure 1**. These, however, are based on safety requirements and should be accordingly increased if recirculation poses a potential problem. Other recommendations for combating hot-air recirculation include:

1. Using induced-draft fans, which force the air away from the bundle.
2. Baffles and/or a stack on top of the bundle for a forced-draft unit (or fan or an induced-draft unit) will also direct the air away from the bundle.
3. A-Frame, V-Frame and vertical bundle arrangements should not be used if recirculation is a potential problem.

Noise Limitations - Noise level is normally measured in dBA. Noise level is mostly affected by the type of speed reducer used and the fan tip speed, where a higher tip speed results in higher noise levels. When noise level is a primary concern, VFDs are preferred over autovvariable-pitch fans, because for a decreased load the VFD will decrease the fan speed resulting in lower noise levels.

Airside Fouling - Normally air-side fouling is not considered in the thermal design of an ACHE. As fins foul the fin efficiency increases, thus reducing the negative effect of fouling. Fouling has a greater effect on the system's air-side hydraulic resistance. API-661 requires that the fan and motor must be capable of 110% of the design air-flows to account for the increased hydraulic resistance added by unaccounted factors such as fouling.

Airside Cleaning - A common way to clean the air-side of ACHEs is to use low-pressure steam. Water spray is also used, but care must be taken to avoid corrosion of the fins. There are vendors who use a chemical foam that contains usually acetic or dilute hydrochloric acid to remove foulant. After spraying the foam, a basic solution such as soda ash is applied to neutralize the acid. Chemical cleaning requires special precautions, and there has been some negative feed back from customers regarding individual vendors and results.

Solar Radiation - The maximum effect of solar radiation is about 200 to 300 Btu/hr ft² (5.5 to 8.2 W/hr) of the bundle face area (length X width) and is normally very small compared to the ACHE heat load.

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INFORMATION REQUIRED FOR DESIGN SPECIFICATIONS

The following information, in addition to the requirements listed on the air-cooled exchanger specification sheet (**Table 6A** or **7A**), should be provided:

1. Air-Side Conditions
 - a. Rainfall, snowfall, hail intensity, frequency, and duration.
 - b. Prevailing wind direction, frequency, and intensity.
2. Process-Side Conditions
 - a. Winterization protection.
 - b. Any planned variation in feed rate or composition which would affect fluid properties such as viscosity, or allowable pressure drop.
3. General Requirements
 - a. Unit elevation (if not to be at grade).
 - b. Orientation of unit with respect to predominant wind direction.
 - c. General description of any other units or building within 100 ft (30 m) of air-cooler (plot plant preferred if available).
 - d. Centerline to centerline distance of piperacks if unit is mounted on piperacks.
 - e. Seismic data and wind load.
 - f. Noise limits.
4. Information from Supplier
 - a. Recommended flow arrangement at 25, 50, 75, and 100% of design throughput at design and minimum ambient temperatures.
 - b. Recommended fan arrangement and blade angle settings for each of the above operating conditions.
 - c. Noise level for proposed unit.
 - d. Estimated man hours for erection or assembly in the field.
 - e. Recommended spare parts list.
 - f. Fan performance curve.

DESIGN PROCEDURES

RECOMMENDED MODELING TOOL

ACE2 from Heat Transfer Research, Inc., (HTRI) is the recommended tool for the simulation and rating of air-cooled heat exchangers. The exchanger may consist of any pass arrangement and motor sizing is included. ACE2 is a validated ACHE model and can be accessed through Hextran. ACE2 is also available as a PC program from ER&E Manuals Distribution. An updated list of ACE2 programming bugs and limitations can be found on the ER&E Heat Transfer Section WebPage under:

<http://159.70.37.160/eedhtran/sradocs/uftech/DMTOOLS/detail.htm>.

See **Reference 5** for information on how to use ACE-2 in Hextran.

SHORT-CUT DESIGN PROCEDURE

A short-cut design procedure is presented in the form of a calculation sheet (**Table 2**). It is intended only for rough sizing and cost estimating purposes. It is not intended for rating detailed designs. The method consists of calculating a temperature efficiency correction and performance factors for given service conditions. The performance factor incorporates an overall heat transfer coefficient generalized by geometric and mechanical considerations. The procedure yields estimates of plot area, fan power requirements, and finned surface area for a typical air-cooled exchanger. It is roughly based on a bundle containing 5 rows of tubes on a 2 5/8 in. by 2 3/8 in. (67 mm by 60 mm) pitch with 2 or 3 tube passes. The tubes are 2 1/4 in. (57 mm) over the fin diameter with a 1 in. (25.4 mm) root diameter, 30 ft (9 m) long with 11 fins/in. (433 fins/m) of 0.016 in. (0.4 mm) thickness and a 23 to 1 finned to bare surface area ratio. The plot area and fan power calculations generally yield values within $\pm 20\%$, and the surface area within $\pm 30\%$ of vendor design data. This method should not be used for fluids with a viscosity greater than 10 cP (0.01 Pa•s). A sample calculation is shown in **Table 3**. This short cut design procedure is available in Pegasys.

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DESIGN PROCEDURES (Cont)

DETAILED DESIGN PROCEDURE

A detailed calculation procedure is also presented in the form of a calculation sheet (**Table 4**). The correlations used were developed or modified by Exxon Engineering from literature data. The basic method used to calculate an air-cooled exchanger is analogous to a shell-and-tube exchanger calculation. A preliminary layout is assumed and then checked via the correlations for the service in question. If necessary, adjustments are made in geometry or air temperature rise in order to balance the assumed and required conditions. A sample calculation is shown in **Table 5**.

Exxon Engineering's position at this time is to duty specify air-cooled exchangers. This is warranted by industry competition and the wide variety of available design configurations. Therefore, the detailed calculation procedure is particularly useful for rating vendor designs wherein the exchanger layout and physical parameters are established. Where physical data are known, this procedure is generally accurate to within $\pm 10\%$.

SAMPLE CALCULATIONS

Sample calculations for both the short-cut (**Table 3**) and detailed design (**Table 5**) procedures are shown for the problem of sizing an ACHE for a Lean Oil cooling service. For comparison of the design capabilities of each method, an Exxon ACHE Data Sheet (**Table 6**) has been completed for an actual vendor design for the same ACHE service (**Table 7**). Comparing the results illustrates how the detailed procedure can yield an adequate design configuration, which is noticeably different from a vendor's configuration. The vendor's choice is based on his standard fin type and tube spacing. It appears for this case that the vendor's choice is a more efficient design from a required fan power standpoint. A comparison of the short-cut method illustrates its general nature in that the results obtained are representative of either the detailed method configuration or the vendor's configuration.

NOMENCLATURE

AAV	=	Actual air volume flow rate, ft ³ /min (m ³ /s)
A _f	=	Area of fins per unit tube length, ft ² /ft (m ² /m)
A _{FF}	=	Free flow area of exchanger, ft ² (m ²)
A _i	=	Inside area of tube, ft ² /ft (m ² /m)
A _o	=	Total outside area of tube per unit tube length, ft ² /ft (m ² /m)
A _p	=	Projected finned area of tube per unit tube length, ft ² /ft (m ² /m)
A _{PL}	=	Plot area, ft ² (m ²)
A _r	=	Area of tube at base of fins, ft ² /ft (m ² /m)
A _S	=	Exchanger surface area (based on outside area), ft ² (m ²)
A _u	=	Unfinned tube area per unit tube length = A _o - A _f , ft ² /ft (m ² /m)
c _p	=	Heat capacity of air (specific heat), Btu/lb-°F (kJ/kg-K)
D _i	=	Inside diameter of tube, in. (mm)
D _o	=	Outside diameter of tube, in. (mm)
D _{of}	=	Fin outer diameter, in. (mm)
D _r	=	Tube diameter at base of fins (root diameter), in. (mm)
E _D	=	Driver efficiency
E _f	=	Fin efficiency
E _{FN}	=	Fan efficiency
E _t	=	Temperature efficiency, short-cut procedure
E _w	=	Weighted fin efficiency
f	=	Friction pressure loss factor, nondimensional
F _A	=	Face area of bundle (length x width), ft ² (m ²)
F _{BH}	=	Heat transfer bundle constant based upon tube geometry, in. (mm)
F _{BP}	=	Pressure drop bundle constant based upon tube geometry, dimensionless
F _n	=	LMTD correction factor
F _t	=	Temperature correction factor, short-cut procedure
G	=	Mass velocity of air at minimum area, lb/hr-ft ² (kg/s-m ²)
H	=	Fin height, in. (mm) = (D _{of} - D _r) / 2
h _{io}	=	Inside film coefficient based on outside bare tube area = 1/R _{io} , Btu/hr-ft ² -°F (W/m ² -K)



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NOMENCLATURE (Cont)

h_o	=	Outside film coefficient based upon finned area = $1/R_o$, Btu/hr-ft ² -°F (W/m ² -K)
j	=	Abscissa LMTD correction factor
k	=	Thermal conductivity of air, Btu/hr-ft-°F (W/m-K)
k_f	=	Thermal conductivity of fin metal, Btu/hr-ft-°F (W/m-K)
k_w	=	Thermal conductivity of tube metal at average tube metal temperature, Btu/hr-ft-°F (W/m-K)
L	=	Length of tube, ft (m)
ℓ	=	Tube wall thickness, ft (m)
LMTD	=	Log mean temperature difference, °F (°C)
M	=	Flow rate, lb/hr (kg/s)
N	=	Fin density, fins/in. (fins/m)
N_T	=	Number of tubes per exchanger
P	=	Motor fan power, HP (kW)
PF	=	Performance Factor - short-cut method
Pr	=	Prandtl number, dimensionless
$P_{t(min)}$	=	Motor fan power at minimum ambient, HP (kW)
ΔP	=	Static pressure drop, in. H ₂ O (kPa) or psi (kPa)
Q	=	Total rate of heat transfer, (exchanger heat duty), Btu/hr (W)
R	=	Number of tube rows
Re	=	Reynolds number, dimensionless
r_{io}	=	Inside fouling factor based on outside area, hr-ft ² -°F/Btu (m ² -K/W)
R_{io}	=	Inside film resistance based on outside area, hr-ft ² -°F/Btu (m ² -K/W)
R_o	=	Outside film resistance, hr-ft ² -°F/Btu (m ² -K/W)
R_{SF}	=	Ratio total surface area to face area of bundle, dimensionless
r_w	=	Tube wall resistance based on outside area, hr-ft ² -°F/Btu
S_F	=	Fin pitch, in. (mm)
S_L	=	Longitudinal distance from center of tube in one column to center of nearest tube in next column (spacing of rows), in. (mm)
S_R	=	Row spacing, in. (mm)
S_T	=	Transverse pitch = center-to-center distance from tube to tube in one transverse row, in. (mm)
t_{avg}	=	Average temperature = $(t_1 + t_2) / 2$, °F (°C)
t_f	=	Air temperature at fan; equal to t_1 for forced draft, t_2 for induced draft, °F (°C)
t_{min}	=	Minimum design air inlet temperature, °F (°C)
t_1	=	Bundle air inlet temperature, °F (°C)
T_1	=	Process fluid inlet temperature, °F (°C)
t_2	=	Bundle air outlet temperature, °F (°C)
T_2	=	Process fluid outlet temperature, °F (°C)
Δt_e	=	Effective log mean temperature difference = $F_n \Delta t_{lm}$, °F (°C)
Δt_{lm}	=	Uncorrected log mean temperature difference (LMTD), °F (°C)
U_o	=	Overall heat transfer coefficient, Btu/hr-ft ² -°F (W/m ² -K)
V_F	=	Face velocity of air (at standard conditions), ft/min (m/s)
Y	=	Mean fin thickness in. (mm)
Y_{SH}	=	Heat transfer correlation factor, nondimensional
μ	=	Viscosity of air at average temperature, cP (Pa•s)
μ_w	=	Viscosity of air at tube wall temperature, cP (Pa•s)
ρ_f	=	Fan air density, lb/ft ³ (kg/m ³)
ρ_s	=	Density of air at standard conditions, lb/ft ³ (kg/m ³)
$\rho_{t(avg)}$	=	Average air density, lb/ft ³ (kg/m ³)
$\rho_{t(min)}$	=	Air density at minimum design air temperature, lb/ft ³ (kg/m ³)



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TABLE 1
DESIGN INLET AIR TEMPERATURES⁽¹⁾

LOCATION	MAXIMUM DRY BULB ⁽²⁾		MINIMUM DRY BULB ⁽³⁾	
	°F	°C	°F	°C
Altona	90	32	—	—
Amuay	90 105	32 for coolers 41 for condensers	70	21
Augusta	95	35	33	1
Bataan	90	32	72	22
Baton Rouge	90	32	20	-7
Baytown	95	35	32	0
Bayway	90	32	33	1
Benicia	85	29	31	-1
Billings	91	33	-11	-24
Campana	90	32	25	-4
Edmonton	71 78	22 noncritical 26 critical ⁽⁴⁾	-28	-33
Fawley	75	24	16	-9
Halifax	75	24	8	-13
Ingolstadt	85	29	3	-16
Kawasaki	92	33	32	0
Koln	86	30	-3	-19
Managua	93	34	66	19
Malaysia (Port Dickson)	91	33	—	—
Okinawa	89	32	41	5
Port Jerome	85	29	10	-12
Rotterdam	76	24	15	-9
Sakai	—	—	—	—
Sarnia	85	29	0	-18
Singapore	92	33	72	22
Socabu	85	29	10	-12
Stenungsund	70	21	8	-13
Thailand (Sriracha)	92	33	62	17
Trecate	85	29	27	-3
Wakayama	89	32	40	4

Notes:

- (1) Values shown should only be used for estimation work. For Design Specification work, the values should be checked against current meteorological data.
- (2) Values shown do not include any allowance for recirculation. A minimum 5°F (3°C) should be added to these values for recirculation. However, the exact allowance depends on location and critical nature of the service.
- (3) For motor sizing.
- (4) Critical services are defined as services where tower or process operation would be difficult to operate and control if air temperature exceeded design air temperature for more than two hours.

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TABLE 2
SHORT-CUT CALCULATION FORM FOR AIR-COOLED HEAT EXCHANGERS

Exchanger No. E - _____

Refinery _____

Service _____

Calc By/Date _____

	Customary Units	Metric Units
1. Exchanger Duty, Q	= _____ Btu/hr	= _____ W
2. Process Fluid Temperatures (T_1 , T_2) and Air Inlet Temperature (t_1),		
T_1	= _____ °F	= _____ °C
T_2	= _____ °F	= _____ °C
t_1	= _____ °F	= _____ °C
3. Temperature Efficiency, $E_t = \frac{T_1 - T_2}{T_1 - t_1}$		
4. Temperature Correction Factor		
$F_t = 1.0 - 0.391 E_t - 0.242 E_t^2 =$ _____		
5. From Table 8 , Performance Factor		
PF	= _____	= _____
6. Plot Area, $A_{PL} = \frac{Q}{PF F_t (T_1 - t_1)}$	= _____ ft ²	= _____ m ²
7. Power, P	= $A_{PL} / 10.76$	= $(0.746) A_{PL}$
	= _____ HP	= _____ kW
8. Finned Surface Area		
$A_S = 125 A_{PL}$	= _____ ft ²	= _____ m ²



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TABLE 3
SHORT-CUT CALCULATION FORM FOR AIR-COOLED HEAT EXCHANGERS (SAMPLE)

Exchanger No. E - _____
Service _____

Refinery _____
Calc By/Date _____

	Customary Units		Metric Units
1. Exchanger Duty, Q	= <u>9.5 x 10⁶</u> Btu/hr		= <u>2.78 x 10⁶</u> W
2. Process Fluid Temperatures (T ₁ , T ₂) and Air Inlet Temperature (t ₁),			
T ₁	= <u>370</u> °F		= <u>188</u> °C
T ₂	= <u>130</u> °F		= <u>54</u> °C
t ₁	= <u>95</u> °F		= <u>35</u> °C
3. Temperature Efficiency, E _t = $\frac{T_1 - T_2}{T_1 - t_1}$			
	= <u>0.87</u>		
4. Temperature Correction Factor			
F _t = 1.0 - 0.391 E _t - 0.242 E _t ² =	<u>0.477</u>		
5. From Table 8 , Performance Factor			
PF	= <u>379</u>		= <u>2155</u>
6. Plot Area, A _{PL} = $\frac{Q}{PF F_t (T_1 - t_1)}$	= <u>191</u> ft ²		= <u>17.7</u> m ²
7. Power, P	= A _{PL} / 10.76		= (0.746) A _{PL}
	= <u>17.8</u> HP		= <u>13.2</u> kW
8. Finned Surface Area			
A _S = 125 A _{PL}	= <u>23875</u> ft ²		= <u>2213</u> m ²

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TABLE 4
DETAILED CALCULATION FORM FOR AIR-COOLED HEAT EXCHANGERS

Exchanger No. E - _____
Service _____

Refinery _____
Calc By/Date _____

A. AIR-SIDE CALCULATION

	Customary Units	Metric Units
1. Exchanger Duty, Q	= _____ Btu/hr	= _____ W
2. Process Fluid Temperatures		
Inlet temperature, T ₁	= _____ °F	= _____ °C
Outlet temperature, T ₂	= _____ °F	= _____ °C
3. Air Temperature		
Minimum ambient temperature, t _{min}	= _____ °F	= _____ °C
Inlet temperature, t ₁	= _____ °F	= _____ °C
Assumed Outlet temperature, t ₂ (see Note 1)	= _____ °F	= _____ °C
4. Assumed Overall Coefficient, U _o (Table 9 , fin tubes)	= _____ $\frac{\text{Btu}}{\text{hr} \cdot \text{ft}^2 \cdot ^\circ \text{F}}$	= _____ $\frac{\text{W}}{\text{m}^2 \cdot \text{K}}$
5. Assumed Number of Tube Rows, R (see Note 2)	= _____	= _____
6. Effective Log Mean Temp. Difference, Δt _e		
a. LMTD, Δt _{l/m}		
ΔT ₁ = (T ₁ - t ₂)	= _____ °F	= _____ °C
ΔT ₂ = (T ₂ - t ₁)	= _____ °F	= _____ °C
$\Delta t_{l/m} = \frac{\Delta T_1 - \Delta T_2}{\ln \left(\frac{\Delta T_1}{\Delta T_2} \right)}$	= _____ °F	= _____ °C
b. Crossflow correction factor, F _n		
$r = \frac{T_1 - T_2}{t_2 - t_1}$	= _____	= _____

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TABLE 4 (Cont)

$$j = \frac{t_2 - t_1}{T_1 - t_1} = \underline{\hspace{2cm}} = \underline{\hspace{2cm}}$$

$$\text{*Factor } F_n \text{ (Figure 8)} = \underline{\hspace{2cm}} = \underline{\hspace{2cm}}$$

c. Effective temperature difference

$$\Delta t_e = F_n \Delta t_{lm} = \underline{\hspace{2cm}} \text{ } ^\circ\text{F} = \underline{\hspace{2cm}} \text{ } ^\circ\text{C}$$

7. Trial Area, A_S

$$A_S = \frac{Q}{U_o \Delta t_e} = \underline{\hspace{2cm}} \text{ ft}^2 = \underline{\hspace{2cm}} \text{ m}^2$$

8. Face Area, F_A

a. Total surface per unit face area R_{SF} , (Table 10) $= \underline{\hspace{2cm}} = \underline{\hspace{2cm}}$

b. $F_A = \frac{A_S}{(R)R_{SF}} = \underline{\hspace{2cm}} \text{ ft}^2 = \underline{\hspace{2cm}} \text{ m}^2$

9. General Layout

a. Length of bundle $= \underline{\hspace{2cm}} \text{ ft} = \underline{\hspace{2cm}} \text{ m}$

b. Width of bundle $= \underline{\hspace{2cm}} \text{ ft} = \underline{\hspace{2cm}} \text{ m}$

c. No. bundles in parallel $= \underline{\hspace{2cm}} = \underline{\hspace{2cm}}$

d. No. bundles in series $= \underline{\hspace{2cm}} = \underline{\hspace{2cm}}$

e. Overall length of unit $= \underline{\hspace{2cm}} \text{ ft} = \underline{\hspace{2cm}} \text{ m}$

f. Overall width of unit $= \underline{\hspace{2cm}} \text{ ft} = \underline{\hspace{2cm}} \text{ m}$

(Arrange the layout to fit standard size bundles, i.e., Table 11 for extruded fin tables)



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TABLE 4 (Cont)

10. Physical Properties of Air

- a. Average temperature

$$t_{avg} = \frac{t_2 + t_1}{2} = \underline{\hspace{2cm}} \text{ } ^\circ\text{F} = \underline{\hspace{2cm}} \text{ } ^\circ\text{C}$$

- b. From **Figure 9**,

$$\text{Specific heat, } c_p = \underline{\hspace{2cm}} \frac{\text{Btu}}{\text{lb} \cdot ^\circ\text{F}} = \underline{\hspace{2cm}} \frac{\text{kJ}}{\text{kg} \cdot \text{K}}$$

$$\text{Viscosity, } \mu = \underline{\hspace{2cm}} \text{ cP} = \underline{\hspace{2cm}} \text{ Pa}\cdot\text{s}$$

$$\text{Prandtl factor, } k \text{ Pr}^{1/3} = \underline{\hspace{2cm}} \frac{\text{Btu}}{\text{hr} \cdot \text{ft} \cdot ^\circ\text{F}} = \underline{\hspace{2cm}} \frac{\text{W}}{\text{m} \cdot \text{K}}$$

- c. From **Figure 10**,
Density ratio @ t_{avg}

$$\rho_{t(avg)} = \frac{0.075 \rho_{t(avg)}}{\rho_s} = \underline{\hspace{2cm}} \frac{\text{lb}}{\text{ft}^3} = \underline{\hspace{2cm}} \frac{\text{kg}}{\text{m}^3}$$

11. Quantity of Air Required, M

$$M = \frac{Q}{c_p(t_2 - t_1)} = \underline{\hspace{2cm}} \text{ lb/hr} = \underline{\hspace{2cm}} \text{ kg/s}$$

12. Free Flow Area of Unit (Total), A_{FF}

a. Tube length, L = $\underline{\hspace{2cm}}$ ft = $\underline{\hspace{2cm}}$ m

- b. From **Table 10**,

$$\text{Total outside area per ft, (m) of tube, } A_o = \underline{\hspace{2cm}} \text{ ft}^2/\text{ft} = \underline{\hspace{2cm}} \text{ m}^2/\text{m}$$

$$\text{Projected fin area per ft, (m) of tube, } A_p = \underline{\hspace{2cm}} \text{ ft}^2/\text{ft} = \underline{\hspace{2cm}} \text{ m}^2/\text{m}$$

- c. Number of tubes

$$N_T = \frac{A}{A_o L} = \underline{\hspace{2cm}} = \underline{\hspace{2cm}}$$

d. $A_{FF} = F_A - A_p L \frac{N_T}{R} = \underline{\hspace{2cm}} \text{ ft}^2 = \underline{\hspace{2cm}} \text{ m}^2$